THE MODERN STEAM TURBINE

THE MODERN STEAM TURBINE

By

E. A. Kraft, Dr.-Ing., Dr. techn. h. c.
Honorary Professor at the Charlottenburg Technical University
Director of the A.E.G.-Turbine Works
Berlin

With 250 illustrations

1931
VDI-VERLAG G.M.B.H. / BERLIN NW 7

Preface

About ten years ago the construction of steam turbines began to make rapid headway. The movement has since been encouraged by the new and unsettled economic conditions, and great progress has been, and still is being, made. Hence it might seem, at first sight, as if the steam turbine were as far away as ever from any standardization, such as has been the culmination of the development of so many other branches of engineering. However, the widely divergent designs of turbines, built in many different countries, have all been constructed with the same end in view. The turbine must be designed in the best possible manner to suit the special economical and working conditions of the site. There are many ways of achieving this end, old and new, practical and fanciful; some are temporary innovations, others have been long well known. If these attempts are examined it will be seen that, in every one of the numerous solutions, there are some principles which remain unchanged in spite of any deviation in the direction of the development.

The object of my book is to make these rules clear to an outsider, to guide him and to give him reliable advice so that he may distinguish the useful from the worthless. I shall not deal with either the theory or the general design of the turbine, as these have been described in such a masterly manner in *Stodola*'s classical work. I shall confine myself to the latest tendencies in turbine construction and cannot, therefore, claim to have exhausted the subject. There are no complicated formulae or equations in this book, but a critical survey, founded on practical experience, is made of the most important problems of the day in the turbine industry. A great number of examples of actual turbines are given which show the application of the latest tendencies. They correspond to the most modern ideas and to the present means at our disposal. In the first German edition (1926) of this book I gave the guiding principles which still remain valid; the newer machines, which I have described in the present edition, show further perfections and refinements. As was to be expected, however, no fundamental changes have taken place.

I have added to this edition short chapters on governing, on turbines of special design, on turbines for very high pressures, and on condensers and their auxiliaries. The question of the materials for steam turbine construction is so very important that the chapter on the subject has been extensively enlarged. All chapters are brought up to date and the obsolete has been left out. In order to arrange the matter more concisely and clearly a few rearrangements have been made.

Illustrations of turbine sections and detail drawings have been shown as clearly as possible, for secret designing is not the true way to progress. I have been glad to find that this view is also held by all the important turbine firms of Europe and America, and I have been readily supplied with drawings and practical information. If it is found that I draw perhaps too many of my examples from designs with which I have been connected, this should not be

taken for partiality. I have, naturally, more information at hand about my ov designs than about those of other builders.

I wish to express my gratitude to all friends and firms who have helped m either in collecting data or in preparing and correcting this edition.

May this present English edition be accompanied by the wish that t reader, be he designer, purchaser or scientist, may acquire a deeper knowled of the latest problems of steam turbine construction and a clearer understandi of the practical values of the new ideas. May it also encourage the turbi designer to further progress.

Berlin, March, 1930

E. A. Kraft

Contents

page

Introduction 1

I. Methods of attaining higher economy 3

1. Preparation of steam turbine schemes 3
   - a. Live steam pressure and temperature 3
   - b. Back-pressure 6
   - c. Interstage reheating 13
   - d. Regenerative feed-water heating 16
   - e. Power generation from process steam 18
2. Design 23
   - a. Steam path 23
   - b. Quality figure 31
   - c. Method of expansion 33
   - d. Nozzles 34
   - e. Moving blades 39
   - f. Casings 46
   - g. Glands 54
   - h. Bearings 58
   - i. Rotors 64
   - j. Governing 67

II. Technical factors limiting the design 77

1. Properties of the materials 77
   - a. Cast iron 77
   - b. Cast steel 79
   - c. Mild steel 80
   - d. Special steels 83
   - e. Non-ferrous metals 87
2. Testing of the materials 87
   - a. Methods of testing 87
   - b. Test pieces 91
3. Stresses 93

III. Principles of design 98

IV. Practical applications of the principles of design 101

1. Turbines for direct drive 101
   - a. High-pressure turbines 101
   - b. Back-pressure turbines 131
   - c. Extraction turbines 138
   - d. Low-pressure and mixed-pressure turbines 145
   - e. Special designs 149
2. Turbines for indirect drive 155
3. Turbines for very high pressures 173

V. Condensers and their auxiliaries 186

Conclusion 198

Index 199

Abbreviations

A. E. G.	Allgemeine Elektricitäts-Gesellschaft, A.-G., Berlin (Germany)
Allis-Chalmers . .	Allis-Chalmers Manufacturing Co., Milwaukee, Wis. (U. S. A.)
B. B. C.	Brown, Boveri & Cie. A.-G., Baden (Switzerland), and Mannheim (Germany)
Bergmann . . .	Bergmann-Elektricitäts-Werke A.-G., Berlin (Germany)
B. T. H.	The British Thomson-Houston Co., Ltd., Rugby (England)
Elliott	Elliott Co., Jeanette, Pa. (U. S. A.)
English Electric .	The English Electric Co., Ltd., Rugby (England)
Erste Brünner . .	Erste Brünner Maschinen-Fabriks-Gesellschaft, Brünn (Czechoslowakia)
Escher Wyss . .	A.-G. Escher Wyss & Cie., Zurich (Switzerland)
Amer. G. E. C. . .	General Electric Co., Schenectady, N. Y., and West Lynn, Mass. (U. S. A.)
Brit. G. E. C. . .	The General Electric Co., Ltd., Erith, Kent (England)
de Laval	A.-B. de Lavals Angturbin, Stockholm (Sweden)
Ljungström . . .	A.-B. Ljungström Angturbin, Stockholm (Sweden)
M. A. N.	Maschinenfabrik Augsburg-Nürnberg A.-G., Nuremberg (Germany)
Metro-Vick . . .	Metropolitan-Vickers Electrical Co., Ltd., Manchester (England)
Oerlikon	Maschinenfabrik Oerlikon, Oerlikon (Switzerland)
Parsons	C. A. Parsons & Co., Ltd., Newcastle-on-Tyne (England)
S. S. W.	Siemens-Schuckertwerke A.-G., Berlin (Germany)
Stork	Gebr. Stork & Co., Hengelo (Holland)
Terry	The Terry Steam Turbine Co., Hartford, Conn. (U. S. A.)
Weir	G. & J. Weir, Ltd., Cathcart, Glasgow (England)
Westinghouse . .	Westinghouse Electric & Manufacturing Co., Philadelphia Pa. (U. S. A.)
Wumag	Waggon- u. Maschinenbau A.-G., Görlitz (Germany)
Yarrow	Yarrow & Co., Ltd., Scotstoun, Glasgow (England)

Introduction

Any innovation in Engineering will remain unnoticed and worthless if it does not involve an economical gain. The direction of the development of a technical discovery, such as a machine, is determined by the economic conditions of the time. Certain requirements are strongly emphasized, while others, which in their time may have been just as important, are put aside. The steam turbine has, under the unstable modern economic conditions, undergone many changes. In considering its development in the present book, we should like to set one principle at the head of our work. There is *one* requirement which must always retain its great importance, independently of any change in outlook or difference of opinion: *The demand for absolute reliability.* It is indisputable that its importance has even increased, due to greater adoption of high steam pressures and temperatures. Formerly, a cheap plant, occupying little space, was the most important condition after that of reliability.

Of late years, fuel being scarce and expensive, the second most important condition has become temporarily the demand for a small steam and fuel consumption. Great improvements have undoubtedly been made. It is, however, unfortunately too often overlooked that a good steam consumption does not necessarily mean a good plant efficiency. The materials required, the weight, the consumption in oil and other substances required when running, the expenses for supervision and standing-by, the space occupied, and last but not least, the capital outlay, are important for determining the economy. A small space requirement means cheap foundations and building; a small weight means reduced loading, transport and erection costs; a low first cost means a moderate expense for interest and amortization of the capital; a simple plant means cheap maintenance.

The losses from even a short close-down can be very great, even if the expenses of the urgent repairs are not considered. They can be particularly great if they occur during a time when the demand for current is heavy and the power station is working at a high load. If it is realized that an involuntary stoppage may easily mean the loss of many months' economy in fuel, then it will be seen that reliability is not only primarily important from a technical point of view, but perhaps even more so from an economic point of view. Then the great weight which experienced designers and responsible managers of power stations attach to this point will be understood.

It will not be in contradiction with this general conception of economy when, in what follows, we shall frequently adopt, to avoid clumsy expressions, the word "economy" in a narrower sense. We want to make it clear that in the first chapter we shall always mean by "economy", an economical steam and fuel consumption, or a high thermodynamic and thermal efficiency of the steam turbine plant. Naturally, the commercial efficiency is more important than either the technical, thermodynamic or thermal efficiencies. This, however, can only be mentioned in the present book, which is exclusively devoted to the engineering side of steam turbine construction.

In surveying the recent developments of steam turbines we see that the endeavour to improve the heat utilization in power stations by raising the steam

pressure and temperature, by a regenerative or by a back-pressure cycle or by other methods, has had a permanent influence on steam turbine plants. By means of theoretical research, and by improved design, the efficiency of the steam turbine itself has been improved at the same time and has now nearly reached the limit which modern materials impose. These economic tendencies are not confined to any country, but they are now general over the whole world. Under changed economic conditions the useful progress in steam turbine construction, based on the greatest saving of fuel, will not remain stationary, but will proceed still further.

The present book is intended to give a brief survey of the modern tendencies in steam turbine construction due to increased knowledge and experience. At the end of the book various designs, which prove the above principles, will be described.

I. Methods of attaining higher economy

The efficiency of steam turbine plants can be improved in two ways, if only such ways as influence the turbine itself are considered.

Among the methods contained in the first group are the increasing of the available heat drop, by raising the live steam conditions or reducing the pressure of the exhaust steam or by interstage steam reheating, which is a necessary feature for high-pressure plants, and the reduction or complete elimination of the heat loss in the condenser by regenerative or back-pressure cycle.

The second group comprises all the innovations and improvements which affect the form of the turbine, namely its general lay-out, its design from thermodynamic point of view and lastly the construction of its elements.

1. Preparation of steam turbine schemes

a. Live steam pressure and temperature

The available heat drop is determined by the pressure and temperature of the live steam and the exhaust pressure. Theoretically, each of these three factors — assuming the steam is superheated — is independent of the other two. In practice, however, they depend on each other owing to the limited strength of the materials and also to the wetness of the steam when expanding to a high vacuum. Let us now consider the effects of each of these three quantities separately. In each case we shall assume that the other two are constant. We shall obtain thus some well-known results, which are easily seen from the steam tables. We shall repeat them here briefly for the sake of completeness.

Let us suppose that the inlet temperature and vacuum of a condensing turbine are constant. Fig. 1 shows that, as the pressure increases, the total heat

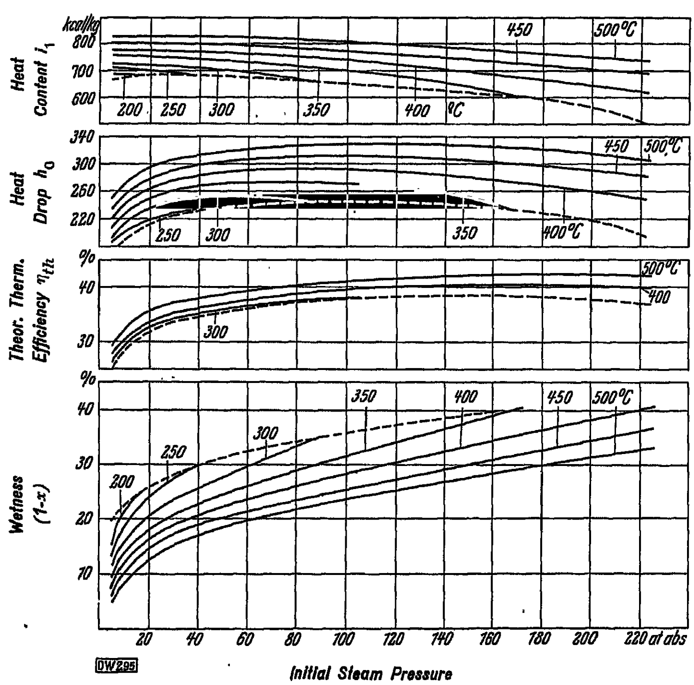


Fig. 1. Adiabatic steam expansion from various initial pressures down to 0.57 lb./sq. in. (0.04 kg./cm.²) absolute for various initial steam temperatures

------- Saturation line

content of the live steam decreases; whilst the available heat drop, for an adiabatic expansion down to the given vacuum, increases at first and only commences to decrease from about 1400 lb./sq. in. (100 kg./cm.²) onwards. Thus, the ratio of this heat drop to the total heat content, which is the theoretical thermal efficiency of the working cycle, increases rapidly at first, then more slowly and finally, at high live steam pressures, begins slightly to fall. At the same time the wetness of the exhaust steam increases with higher initial pressures, as shown on the bottom group of curves.

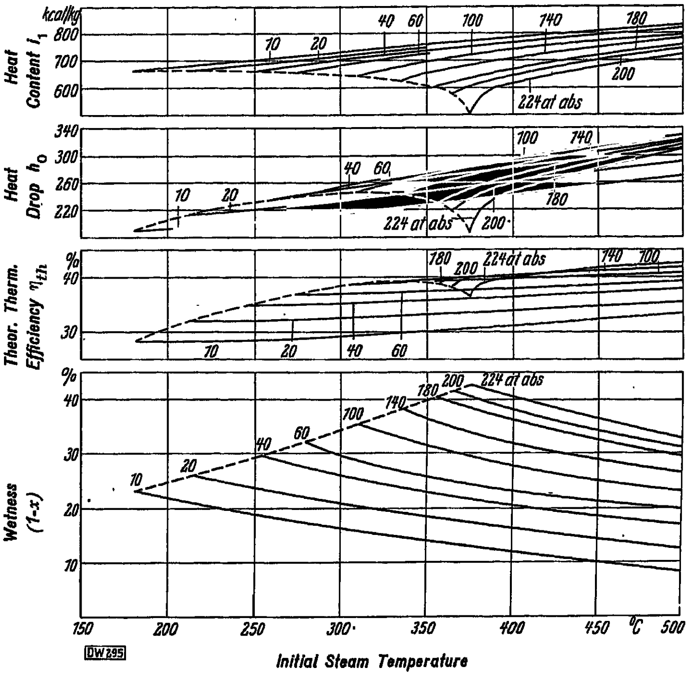


Fig. 2. Adiabatic steam expansion from various initial temperatures down to 0.57 lb./sq. in. (0.04 kg./cm.²) absolute for various initial steam pressures

-------- Saturation line

With increasing temperature of the live steam, at constant initial pressure and vacuum, the heat content, the heat drop and the thermal efficiency increase, as shown in Fig. 2, whilst the wetness of the exhaust steam diminishes. Thus the adoption of higher steam temperatures is always advantageous from a thermodynamic point of view. However, the reduced strength of materials at high temperature fixes a limit to the steam temperature which, for modern practice, is about 840° to, at the most, 930° F. (450 to 500° C.). As an experiment even higher temperatures (1) are used so that, in the near future, it is possible that the limit may be raised.

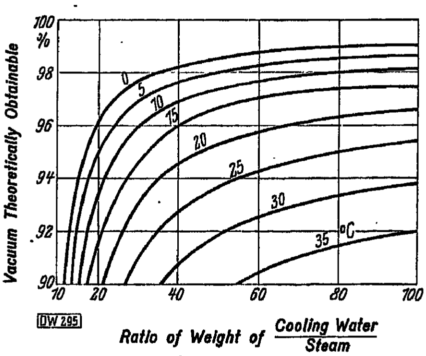


Fig. 3. Vacuum theoretically obtainable with various inlet temperatures of cooling water, for various quantities of cooling water

Finally, when considering the third factor which determines the heat drop, the back-pressure of the turbine, as the independent variable, we find, naturally, that the heat drop increases as the back-pressure decreases. The largest heat drop is obtained by exhausting into vacuum. The practical limit is the vacuum which it is economical to obtain. This, under normal European conditions, is at the most 96 to 97%. The curves in Fig. 3 showing the theoretically obtainable vacuum are based on "no air" condition and a condensate temperature equal to that of the outgoing cooling water.

These purely theoretical deductions from the steam charts cannot, of course, be directly

(1) Refer for instance to the high-temperature turbine for Delray Station, Detroit, U.S.A., described on page 180.

applied either to the thermodynamic efficiency or to the overall economy of the turbine. These are also affected by other conditions which are inseparable from the variations of the quantities fixing the heat drop and depend usually on the physical state of the steam. These relations will now be discussed. In order to do this it is important to consider separately condensing and back-pressure turbines. Condensing turbines are the best known and also the most important type of turbine as regards both number and capacity. They will, therefore, be considered first.

In examining the influence of high steam pressures and large heat drops on the efficiency of condensing turbines, the most important thing is the way in which the heat drop is divided amongst the different parts of the turbine. A fixed division can, naturally, not be given, and in fact for certain types of construction it can only be imagined. There can, however, be fixed an arbitrary division in two parts, which, for short, may be called H.P. and L.P. parts. The division of the total heat drop between the H.P. and L.P. parts depends very largely on the initial pressure. For an initial pressure of 170 lb./sq. in. (12 kg./cm.²) gauge, a superheat of 200° F. (110° C.) and a vacuum of 96%, the total heat drop is 405 B.Th.U./lb. (225 kcal./kg.). Assuming a definite type of design, as an example, with a pressure before the L.P. part of 14.2 lb./sq. in. (1 kg./cm.²) gauge and without considering any reheat factor for the heat drop in the L.P. part, then the H.P. part has 168.3 B.Th.U./lb. (93.5 kcal./kg.) or 41.5%, and the L.P. part 236.7 B.Th.U./lb. (131.5 kcal./kg.) or 58.5%. This division is changed when the live steam pressure increases, the H.P. part then takes a larger portion of the heat drop; at 285 lb./sq. in. (20 kg./cm.²) gauge and the same superheat, the total heat drop becomes 441 B.Th.U./lb. (245 kcal./kg.) 47.5% of which are used in the H.P. part. At 427 lb./sq. in. (30 kg./cm.²) gauge the H.P. part deals with 52% of the total heat drop. This shifting of the distribution is seen even more strikingly if the ratio of the heat drop in the H.P. part to that in the L.P. part is considered. For the first case this ratio is 71%, for the second 90% and for the third 108%. These calculations are based on a constant weight of steam. If a constant load is taken as a basis, the amount of steam decreases, the pressure falls even lower before the L.P. part and there is a larger increase of the H.P. heat drop.

With an increasing initial pressure the internal efficiency of the H.P. part will be all the more important for the total efficiency according to the increase of its heat drop. The efficiency of the H.P. part determines, however, the state of the steam before the L.P. part and thus the improvement in efficiency of the H.P. part does not influence the total efficiency to its full extent. The steam temperature before the L.P. stages is higher for a bad H.P. efficiency than for a good one. As the adiabatic heat drop increases with the superheat, a bad efficiency in the H.P. part increases the adiabatic and the useful heat drops in the L.P. part; on the other hand, a good H.P. efficiency lowers it, i. e. the reheat factor is greater in the first case than in the second. Besides increasing the heat drop, the higher temperature before the L.P. part due to the bad efficiency of the H.P. part also appreciably improves the thermodynamic efficiency of the L.P. stages. Fig. 4 shows approximately, for an average initial pressure, how the total efficiency $\eta i_{tot.}$ of a condensing turbine is influenced by the H.P. efficiency $\eta i_{H.P.}$. For each 1% improvement in the H.P. efficiency the total efficiency of the turbine rises by only about 0.15 to 0.3%, according to the absolute value of the H.P. efficiency and the ratio of the heat drops in the H.P. and L.P. stages. In the example mentioned above, where for an initial pressure of 427 lb./sq. in. (30 kg./cm.²) gauge the H.P. part has 52% of the total heat drop, a total efficiency of 78% and a H.P. efficiency of 60% may be assumed. By increasing the latter to 70%, or by 17%, the total efficiency is raised by $17 \times 0.21 = 3.57\%$, and becomes $78 \times 1.0357 = 80.8\%$. By a further

14% increase of the H.P. efficiency from 70% to 80%, the total efficiency improves by $14 \times 0.27 = 3.78\%$ and reaches 83.85%. According to other data and methods, slightly different values are obtained, but there is always, as an outstanding feature, the small change in the total efficiency relative to that of the H.P. efficiency. This explains why no remarkably rapid increase in the total efficiency of condensing turbines has been made in the last few years, in spite of the great progress in the design of the H.P. part. From the point of view of turbine efficiency, the importance of higher steam pressures is limited, and a considerable increase in the total efficiency of condensing turbines is not to be expected, unless a fundamental improvement in the efficiency of the L.P. part is effected. This means that methods must be found to guard against the effect of moisture in the steam and also to reduce the leaving loss.

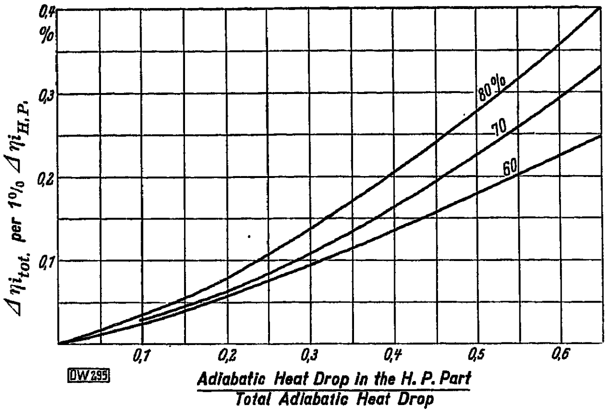


Fig. 4. Increase in the total efficiency $\eta i_{tot.}$ of a condensing turbine due to every 1% increase in the efficiency of the H.P. part above 60, 70 and 80%, in relation to the ratio of the H.P. heat drop to the total heat drop

Calculated for 427 lb./sq. in. (30 kg./cm.²) gauge initial pressure, 750° F. (400° C.) initial temperature $\eta i_{L.P.} = 70$ to 85%

On account of its great density, steam at a high pressure can only be used in a turbine with a good efficiency when the steam and blade velocities are low. This requires a small heat drop per stage and therefore large number of stages. The gain thus obtained is often attributed to the physical properties of low steam velocities. The advantage is, naturally, due more to the greater blade length, with full admission, and also the resulting smaller leakage losses. Due to these considerations, the tendency became to adopt excessively large numbers of stages, and turbines, even of small and medium capacity, were built of the multiple cylinder type. Much disturbance was caused amongst the European turbine builders a few years ago. After having in some cases gone too far, the turbine with a large number of stages is now widely adopted. For small and medium outputs it is now usually endeavoured to use a single casing. For large outputs there are additional reasons why a turbine should be divided into several units. It may be necessary to divide the steam in the last stages if the volume is very large. The maximum unit capacity for generators at a given speed may be surpassed and the load may have to be divided amongst several shafts. The advantages of limiting the number of casings as much as possible, even for large outputs, will be explained fully later. Before deciding on a turbine of a particular design, a careful calculation of the running costs of the boiler and engine room plant should be made. Under the present depressed economic conditions it will usually be found that a single or double-casing machine of good efficiency will really be more economical than a multi-casing machine with only a slightly better maximum efficiency.

b. Back-pressure

If a definite output is to be produced with a given quantity of steam, the state of the steam in the exhaust is determined once the initial pressure and temperature have been chosen. This can easily be seen on the steam chart.

It follows, as shown in Figs. 1 and 2, that with condensing turbines the live steam temperature should be raised at the same time as the pressure. If this is not done the steam would be too wet in the last stages; this would diminish the L.P. efficiency and, consequently, the total efficiency. At the same time the probability of greater wear of the L.P. blades is increased.

Fig. 5 illustrates these relations for the particular case of a condensing turbine. Adiabatic expansion down to 96% vacuum is assumed. The initial steam conditions are plotted which should be chosen if a given wetness of the exhaust steam is not to be exceeded. It also shows how the proportion of the expansion in the superheated region to that in the wet region varies with the live steam pressure and temperature. The lower curves give for different values of the wetness in the exhaust the corresponding values of the initial temperature and pressure. For instance, if the wetness of the exhaust steam should not be greater than 20% for 752° F. (400° C.) initial steam temperature, the pressure should not be higher than about 485 lb./sq. in. (34 kg./cm.²) absolute. For 932° F. (500° C.), however, pressures up to nearly twice this value may be used. The upper curves show another important fact. For a constant wetness in the exhaust, the proportion of the expansion which takes place in the superheated region increases with the initial pressure and temperature.

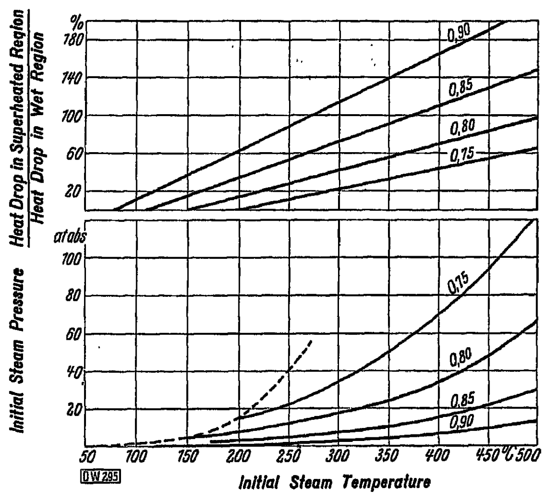


Fig. 5. Initial steam pressure and heat drop for degrees of dryness x = 0.75, 0.80, 0.85 and 0.90 at the end of an adiabatic expansion down to 0.57 lb./sq. in. (0.04 kg./cm.²) absolute, in relation to initial steam temperature

------- Saturation line
(1 − x) = Wetness

If the conditions of the steam flow remain the same, the efficiency of a turbine decreases as the part of the expansion in the wet region increases. This is due to two causes. Firstly, the small drops of water, which are formed, must be accelerated by the steam; work is thus consumed and the flowing mixture of steam and water has a smaller kinetic energy than corresponds to the heat drop; this loss is converted into heat. Secondly, the drops of water, having a much smaller velocity than the steam, hit against the back of the moving blades and produce a braking effect. The variation in turbine efficiency with the wetness of the steam was considered in detail for the first time by *Baumann* (2). Of recent years English and American engineers especially have conducted experiments on this subject (3). It is often assumed that the stage efficiency can be obtained approximately by multiplying the efficiency for superheated or dry steam by the dryness. Although this very simple approximation ignores many factors, it has been verified by tests on L.P. turbines with superheated steam and has proved itself to be very serviceable. It seems to be far better than the previous formulae, which were either empirical or assumed. It should be mentioned that also the supersaturation of steam is connected with the reduction of efficiency due to increased steam wetness.

(2) Proc. Inst. Electr. Eng. 48 (1912) p. 768.

(3) See *A. L. Mellanby* and *W. Kerr:* "The limiting possibilities in steam plants", Engineering 119 (1925) pp. 301, 334 and 366; *W. E. Blowney* and *G. B. Warren:* "The increase in thermal efficiency due to resuperheating in steam turbines", *G.E.C.* Review 28 (1925) p. 662.

It should be understood that not only a high initial pressure, with a relatively low temperature, but also a high turbine efficiency increases the wetness of the exhaust steam. We have thus two opposed factors. A high efficiency, by means of a flow with the minimum of losses, is aimed at by all theoretical and practical means. When thus approaching an adiabatic expansion the wetness of the steam grows, which impairs the efficiency of the L.P. stages, consequently also the total efficiency, and the expansion has to diverge again from the adiabatic. It results from these considerations that, with the present state of our knowledge, the efficiency of condensing turbines is limited by thermodynamical reasons and cannot be increased, even with the most carefully designed blading (4).

The detrimental effect of water in the last stages of condensing turbines can be seen on blades which have been in service for some time. Fig. 6 shows in a schematic way how the unhomogeneous flow of steam and water should be visualized. The absolute velocity of the steam on leaving the guide vanes, calculated from the available heat drop, is shown in magnitude and direction by c. The entrance angle of the blade is designed for the corresponding relative velocity w. The drops of water, however, will flow with the smaller velocity c_1. They will hit the back of the blades in the direction of their relative velocity w_1. The centrifugal force tends to throw the water outwards, towards the tips of the blades. The effect of their impact on the blade material must be the most noticeable in this region. Many turbine owners have discovered, to their cost, that this is what takes place. In steam turbines which have their last stages working in very wet steam, the blades are eroded after running a short time, even if they are made of the best materials. This always happens in the same way on the back of the entrance edge of the blades, near the periphery (Fig. 7). In most cases, however, this erosion grows more rapidly only at the beginning, then more and more slowly. As long as it remains within moderate limits, it is not dangerous as it occurs where the stresses are only small. The remainder of the blade surface may, even after long service, remain unaffected.

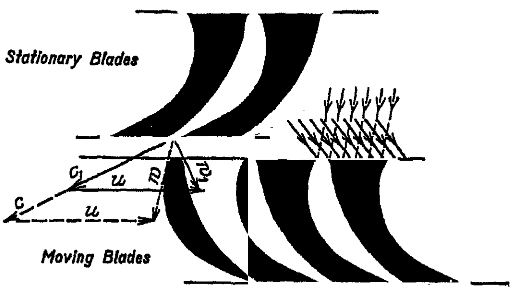


Fig. 6. Effect of moisture in steam on the direction of the relative entrance velocity in condensing turbines

c = Absolute velocity of the steam
c_1 = Absolute velocity of the drops of water
w = Relative velocity of the steam
w_1 = Relative velocity of the drops of water
u = Peripheral velocity

The live steam temperature should, therefore, be chosen so that the steam on leaving the last stage should not, if possible, have a moisture greater than 10%. For thermodynamic reasons, an upper limit of the initial temperature is also fixed for condensing turbines. The steam on leaving the turbine should not be above the saturation point as only latent heat, and not superheat, should be extracted in the condenser. This condition is always fulfilled for high or medium initial pressures, even with high temperatures, provided the load is not too small and the efficiency is normal. Consequently, the highest admissible

(4) See *G. Zerkowitz:* "Zur Frage der Entspannung von Naßdampf in der Dampfturbine", p. 593, "Festschrift *Stodola*" (Zurich: Orell Füssli 1929). This book contains references to the most important publications on the matter. See also *G. Zerkowitz:* "Die Entspannung von Naßdampf in der Dampfturbine", Arch. Wärmewirtschaft 10 (1929) p. 271. *J. von Freudenreich:* "Der schädliche Einfluß der Dampfnässe in Dampfturbinen", *B.B.C.* Mitteilungen (Baden) 14 (1927) p. 119.

steam temperature is a question of materials.

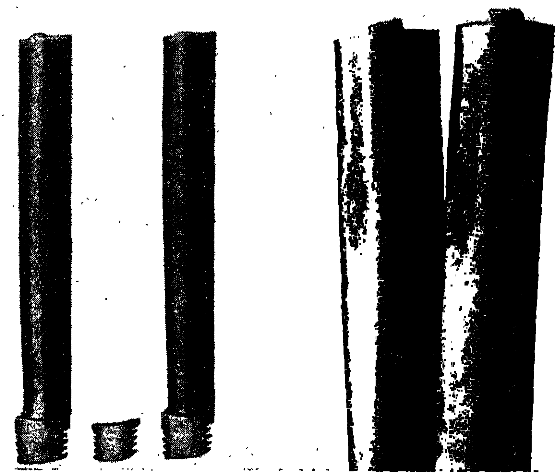
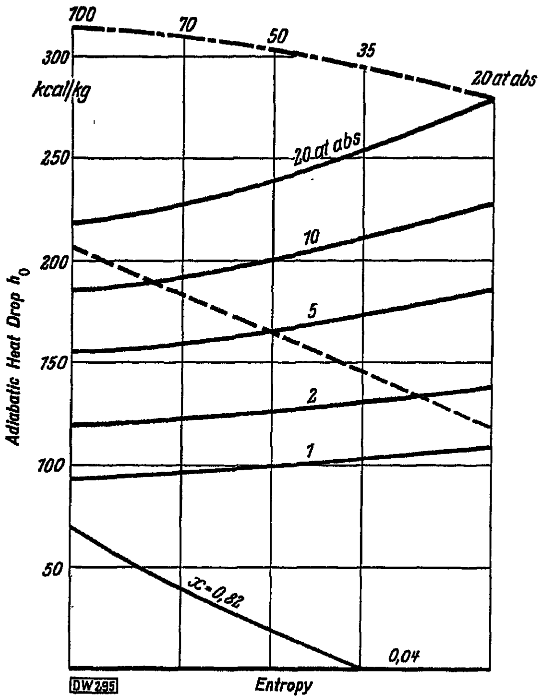


Fig. 7. Effect of moisture in steam on the entrance edges of the tips of moving blades

For back-pressure turbines the conditions are different; since wetness of the exhaust steam is not usually to be feared, the choice of the live steam conditions is freer. Within admissible temperature limit much higher live steam pressures may be used than in the case of condensing turbines. The gain in heat drop which can be obtained by increasing the initial pressure is greater in the case of back-pressure turbines than condensing ones. This is shown in Fig. 8. The gain is the greater the higher the back-pressure. For instance let us suppose a constant initial temperature of 840° F. (450° C.) and that the live steam pressure is initially 285 lb./sq. in. (20 kg./cm.²) absolute. In order to gain 90 B.Th.U./lb. (50 kcal./kg.), the initial pressure must be raised to about 740 lb./sq. in. (52 kg./cm.²) if the back-pressure is 140 lb./sq. in. (10 kg./cm.²) absolute; to nearly 1400 lb./sq. in. (100 kg./cm.²) for 28 lb./sq. in. (2 kg./cm.²) absolute back-pressure; whilst for a condensing turbine a sufficiently high pressure could not be obtained. With back-pressure turbines, therefore, the adoption of high pressures presents greater advantages and possibilities than for condensing machines.

When considering the increase in heat drop which can be realized by lowering the back-pressure, two cases are again to be distinguished. In back-pressure plants, the exhaust pressure depends on the requirements for the process steam. The friction losses in the mains must be taken account of, they depend on the diameter and the length of the pipe line and on the number of bends and valves. If the piping is appropriately dimensioned and laid, and the fittings are used suitably and sparingly, then the back-pressure must be considered as determined and cannot be altered by the designer of the turbine. In condensing turbine plants, the back-pressure, or vacuum in the condenser, depends chiefly on the quantity and temperature of the available cooling water, on the condenser design and sometimes on the effectiveness of cooling arrangement of the circulating water. It is possible, however, to choose these quantities to produce the best effect in the turbine. The exhaust casing must always be constructed so as not to be, so to speak, the narrowest part of the installation, that is, the advantage which is obtained by a good condenser design, or other means, should not be annulled by an unsuitably shaped exhaust casing.

Since the heat drop can be considerably increased by improving the vacuum and the rate of increase grows with the vacuum, the specific steam consumption decreases in a similar manner as long as it is possible to use the extra heat drop with a good efficiency. It is supposed that when ordering a condensing turbine, either the vacuum or the temperature of the cooling water have previously been chosen for which the plant is going to be designed. It is not sufficient simply to take the average of the highest and lowest cooling water temperatures. In most cases the temporary variation of temperature should be taken into account, together with the corresponding number of working days and the expected load.



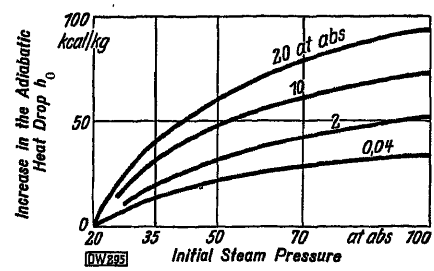


Fig. 8. Adiabatic heat drop for various initial steam pressures and increase in adiabatic heat drop due to an increase in initial steam pressure, for various back-pressures

—·—·— Initial steam pressure
– – – – Saturation line
Initial steam temperature, 840° F. (450° C.)

Since high vacua, to be efficiently utilized, need large areas for both the turbine blading and the exhaust branch, and sometimes also require an increase in the number of turbine stages, the turbine becomes more expensive. This extra cost, however, will usually be worth while if a high vacuum can be attained by lowering the temperature and not increasing the quantity of cooling water. In northern countries, especially by the sea, cold water is available all the year round, the average temperature of which has often worked out at not more than 41 to 43° F. (5 to 6° C.). These stations possess what might be called a free source of power. It is usually economical, however, to design the turbine for the average expected vacuum, and it would be very absurd, apart from the increase in cost, to go to the extreme and dimension the turbine for the lowest cooling water temperature, which may only occasionally occur for short periods. When the last stage of a turbine designed with very large steam passages for dealing with a high vacuum is working with a bad vacuum, the steam volume and heat drop are smaller; the steam velocity is too small in comparison to that of the blade, little power is produced, the wheel so to say churning the steam.

Increased vacuum represents an actual increased heat drop for the turbine when the pressure is not only lower in the condenser, but also immediately after the last row of blades. This means that the last stage must be so designed that the leaving loss is small and that the exhaust branch does not cause any appreciable pressure drop.

The leaving loss of a turbine is unavoidable. Usually the absolute velocity of the steam on leaving the moving blades can be partly or completely utilized in the guide blades of the following stage. This, naturally, cannot be done in the last row. The kinetic energy, which corresponds to the absolute velocity of the steam leaving the last stage, must be considered lost. This loss, for a given steam flow, is larger, the greater the specific volume and the smaller the leaving section.

The top group of curves in Fig. 9 shows the relation between the leaving loss and the pressure after the last stage for different quantities of steam. The outlet ring area of the blading has been chosen as 36.6 sq. ft. (3.4 m.²). The specific steam volume and wetness are based on an assumed expansion. It will be seen that in order to pass large quantities of steam, such as between 150 and 200 tons per hour, through this area, a poor vacuum or a large leaving loss is inevitable.

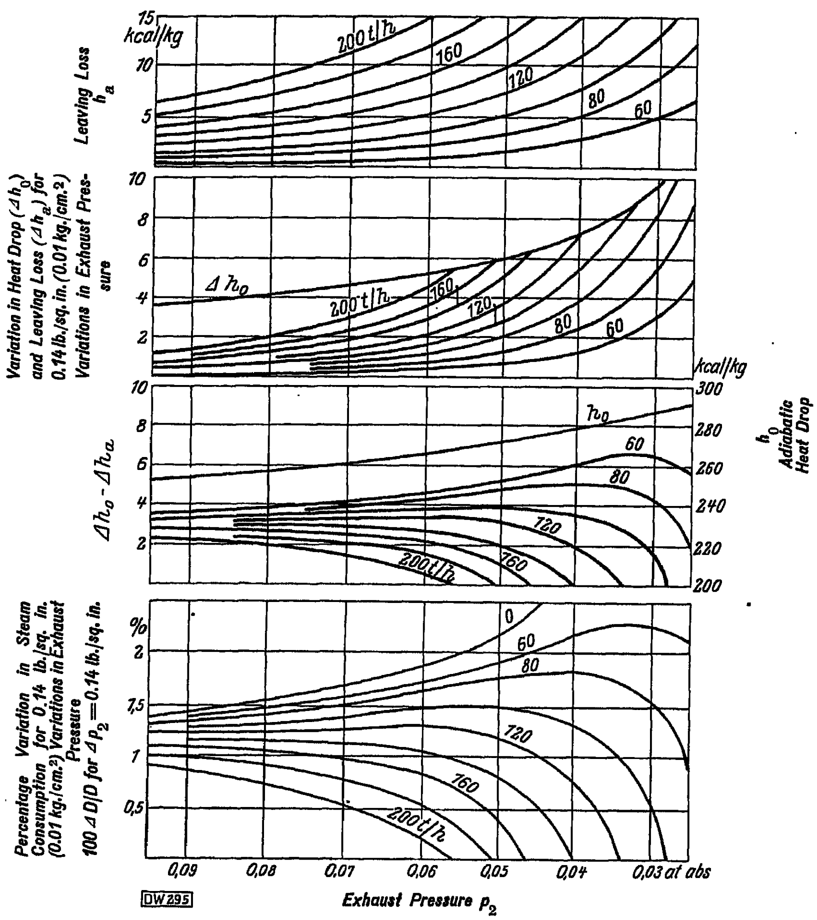


Fig. 9. Leaving loss and vacuum of condensing turbines, for various steam quantities

Initial steam pressure	500 lb./sq. in. (35 kg./cm.²) gauge
Initial steam temperature	750° F. (400° C.)
Outlet ring area	36.6 sq. ft. (3.4 m.²)
Steam flow	G tons/hour
Specific steam consumption	D lb./kw.-h.

The second group of curves shows the increase of the leaving loss for diminutions of 0.14 lb./sq. in. (0.01 kg./cm.²) of the exhaust pressure. The curve Δh_0 gives, for these same falls in pressure, the increase in heat drop along the assumed expansion. The difference between the two gives the third group, showing the useful heat drop which is available when the vacuum is increased by steps of 0.14 lb./sq. in. (0.01 kg./cm.²). If the increase in heat drop is entirely compensated by the increased leaving loss, there will be no gain in useful heat drop or steam consumption. The curve h_0 gives the adiabatic heat drop for the initial conditions of 500 lb./sq. in. (35 kg./cm.²) gauge, 750° F.

(400° C.) and the corresponding back-pressures of 1.42, 1.28, etc., lb./sq. in. (0.10, 0.09, etc., kg./cm.²) absolute. Finally, the lowest group of curves gives, for this same example, the improvement of steam consumption, in per cent of the theoretical steam consumption, for an increasing vacuum.

In modern turbines a leaving loss of more than about 2% is not usually allowed for the economical load. At overload the loss will, naturally, be correspondingly higher.

To diminish the leaving loss, the last stage must have ample steam passages, and consequently, long blades. It will easily be understood that parallel blades, by which are meant blades with the same section for their whole length, can only give a bad efficiency. As can be seen from Fig. 10 the peripheral velocity at the tip is considerably greater than at the root. If therefore, the relative entering speed for a parallel blade is calculated in the usual way with the mean peripheral speed, the steam will strike the inside of the blade with great force at the foot of the blade, while at the tip the steam will be directed on to the back. In order to diminish the loss caused in this way, twisted moving blades are used. At their foot they have a very concave section in order to deviate the jet sharply; at their tip the section is very flat and the steam is much less turned.

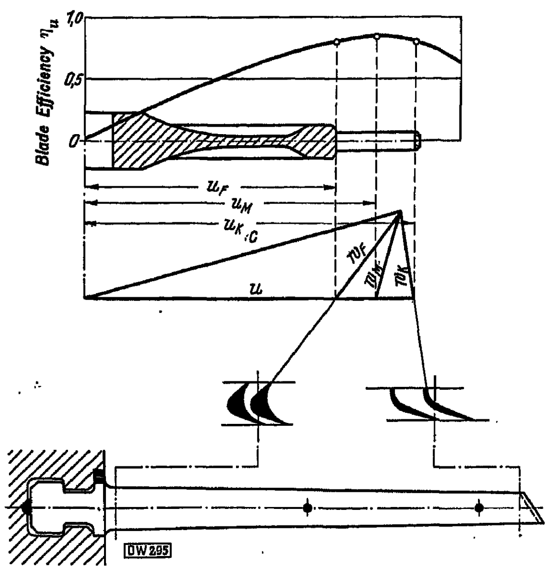


Fig. 10. The reason for twisting long moving blades

c = Absolute velocity of steam
w = Relative velocity of steam
w_F at foot of blade
w_M at mean diameter
w_K at tip of blade
u = Peripheral velocity
u_F at foot of blade
u_M at mean diameter
u_K at tip of blade

In spite of these methods, the velocity of the steam discharging from the last stage will be greater than for the intermediate stages. This particularly applies to the so-called limit turbines, or turbines for a capacity which is just within the limit obtainable for a given speed. As the velocity of the steam leaving the last stage increases, it becomes all the more important to reduce as much as possible the losses in the exhaust casing. All other conditions remaining unchanged, these losses increase with the square of the steam velocity. If it is possible to shape the exhaust branch as an effective diffuser, a large part of the leaving momentum can be reconverted into pressure and a measurable pressure drop in the exhaust casing can be avoided.

By careful improvements in design, manufacture and material of the blading of the last stages, by amply proportioning the exit areas, by giving the exhaust casing the proper stream-line shape, and finally, by dividing the machine into serveral casings, the unit capacity of turbo-generators has constantly increased (Fig. 11). The diagram shown has, like all curves from statistical research, a considerable dispersion of its points in places. It gives, however, a good idea of the tendency of development. The supremacy of America in large turbines should be noted. The curve, also compiled from statistics, which

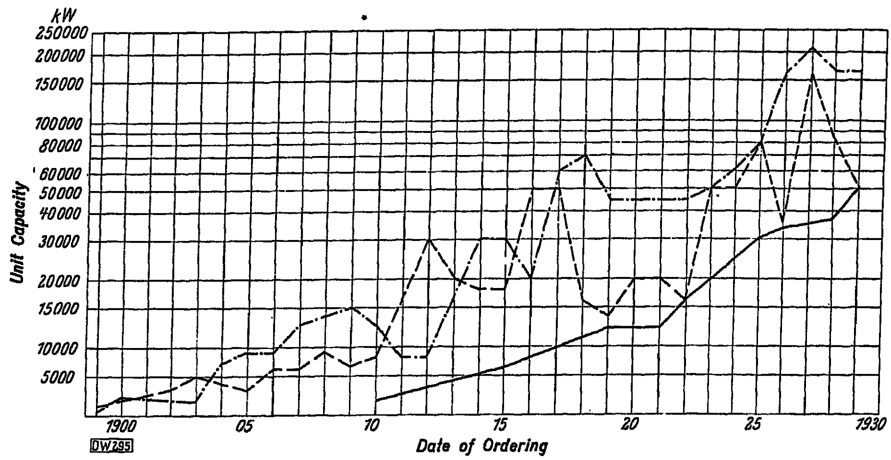


Fig. 11. Increase in the unit capacity of steam turbines

—·—·—·— Largest steam turbine built in America
------- Largest steam turbine built in Europe
——————— Largest 3000 R. P. M. turbine built in Europe

gives the capacity of the largest units at 3000 R.P.M. far exceeds the values for 3600 R.P.M. American turbines. This proves that the greater demand for power is the cause of the larger American units, and not a technical superiority over the European turbine builders. In 1910, machines at 3000 R.P.M. were not built over 1600 kw., in 1920 the limit was already 12,000 kw., in 1925 it was 30,000 kw., at present it is as much as 50,000 kw. on one shaft. Even now developments are not exhausted.

The limit capacities of condensing turbines have, naturally, not increased so rapidly. When considering the limit capacity of a turbine, without concerning oneself with the generator, obviously only outputs should be compared which can be obtained from machines with the same number of exhausts and a permissible leaving loss. The limit capacities for 3000 and 1500 R.P.M. single and double-flow condensing turbines, which were actually built, has been plotted, on Fig. 12, for every year since 1921. We had only the data for the *A.E.G.* when making this curve, those for other firms have not, however, any essential difference in shape, and they all show the tendency towards higher limit capacities.

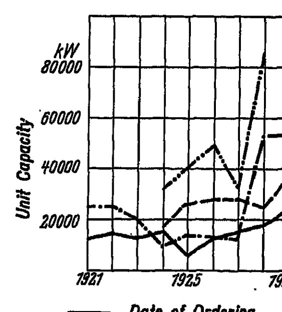


Fig. 12. Limit capacity turbines erected by the *A.E.G.* from 1921 to 1929

——————— 3000 R. P. M., single-flow
------- 3000 R. P. M., double-flow
—·—·— 1500 R. P. M., single-flow
—··—··— 1500 R. P. M., double-flow

c. Interstage reheating

Every steam turbine is part of a working cycle which consists of heating, evaporating and superheating at the constant boiler pressure, then follow an adiabatic expansion in the turbine down to the back-pressure, condensation at constant pressure and temperature, and finally compression to the boiler pressure in the feed pump. This cycle is commonly called, on the continent of Europe, the *Clausius* cycle, whilst English and American writers refer to it as the *Rankine* cycle. If it is compared with the *Carnot* cycle, which has the highest and lowest temperatures for limits, it will be found that the *Clausius-Rankine*

cycle is a relatively poor one. Although previous isolated attempts were made to improve the cycle, it was only when the question of higher steam pressures was tackled that definite proposals, exhaustive theoretical investigations and experimental applications were made. Important additions and improvements to high-pressure power plants are the intermediate reheating of the working steam and also the bleeding of part of the steam from various stages of the turbine for heating the feed-water. The discussion of these two arrangements cannot be separated from that of the whole power plant, they are particularly connected with the boiler, the superheater, the economiser, the feed-heaters and the feed and extraction pumps. The question would take us away from the subject of this book, namely the Steam Turbine. Another reason for only mentioning briefly these two subjects is that recently much has been written about them. We shall only mention the publications of *Blowney* and *Warren* (5), *Collingham* (6), *Guy* (7), *Josse* (8), *Löffler* (9), *Melan* (10), *Mellanby* and *Kerr* (5), *Münzinger* (11), *Noack* (12) and the excellent work of *Zerkowitz* (13).

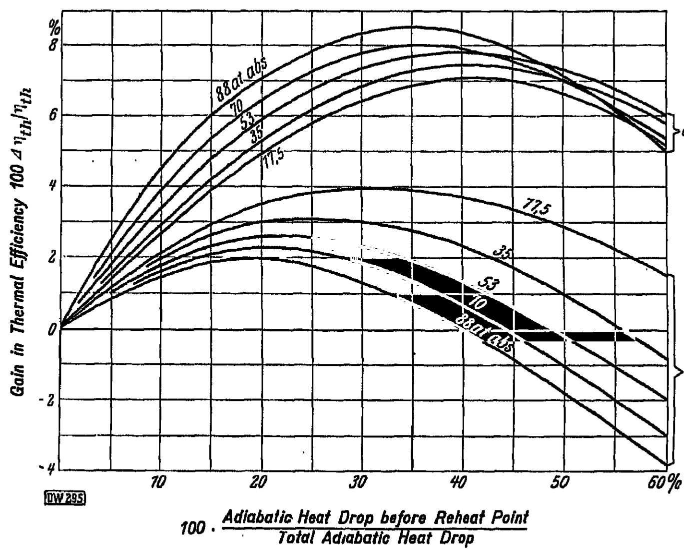


Fig. 13. Gain in thermal efficiency of steam turbines due to reheating

Calculated for
0.50 lb./sq. in. (0.035 kg./cm.²) absolute exhaust pressure
750° F. (400° C.) initial and reheating temperatures
a = Gain due to increase of the heat drop and reduction of the wetness of exhaust steam
b = Gain due to increase of the heat drop only

In the case of reheating, all the steam is resuperheated at constant pressure after it has done work in part of the turbine. It is then returned to the turbine. The operation can be repeated once, or several times. The thermal efficiency of the corresponding cycle is higher than that of the *Clausius-Rankine* cycle. The gain from reheating is due more, however, to the diminution of the wetness in the L.P. part. This is accompanied by an improvement in L.P. efficiency and less wear of the blades from condensed water. The theoretical gain, which is only due to the improved thermal efficiency, is small; for instance, as shown in Fig. 13, 3% is the most that can be obtained with an initial pressure

(5) See the footnote (3) on page 7.
(6) The Electrician 94 (1925) p. 725.
(7) Power Plant Engineering 31 (1927) p. 239.
(8) Zeitschrift des Vereins Deutscher Ingenieure (Z. VdI.) 68 (1924) p. 65.
(9) Z. VdI. 69 (1925) p. 1155.
(10) *H. Melan:* "Die Schaltungsarten von Hilfsturbinen" (Berlin: J. Springer 1926).
(11) Proceedings of the 4th meeting of the Commission on H.P. plants of the "Vereinigung der Elektrizitätswerke" (Germany) (1929) p. 17.
(12) Z. VdI. 67 (1923) p. 1155.
(13) Z. VdI. 68 (1924) p. 147, 1026 & 1093; 73 (1929) p. 1429.

of 500 lb./sq. in. (35 kg./cm.²) absolute. The combined thermal gain due to the increased heat drop and diminished wetness of the exhaust steam can be read from the upper group of curves for various live steam pressures. For our example this gain is 7.5%. These values, however, do not take into account the additional losses that occur, such as those in the piping or the reheater, which will naturally reduce the gain. On the other hand the lengthening of the life of the blades has not been considered either, and this may sometimes compensate the additional losses.

When multiple reheating is investigated, it is found that the gain obtained by more than two reheaters is so small, especially if the throttle losses in the reheaters and piping are considered, that the additional complication ist not profitable.

In practice, there are four ways of arranging resuperheating. Firstly, by reheating in seperately fired superheaters; secondly, by reheaters adjoining the boilers and using the flue gases; thirdly, by using part of the superheat of live

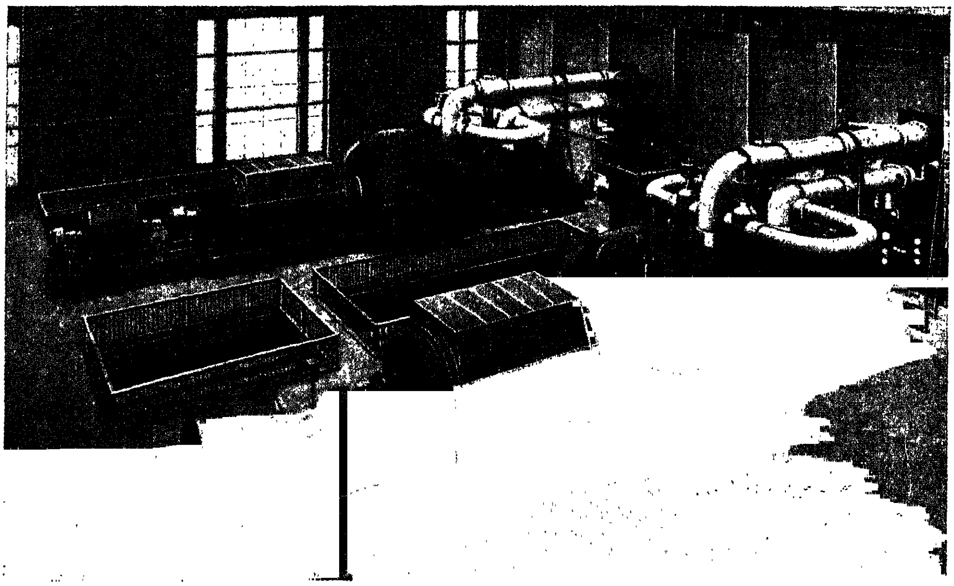


Fig. 14. *Amer. G.E.C.*, turbo-alternators in the Columbia Park Station (equipped with reheating)

steam and finally by condensing live steam. The first method is probably the least economical one. When reheating by means of the flue gases, the steam has to be led from the turbine to the boilers and back, involving unavoidable pressure and temperature drops. A few plants of this kind have been built in America. They show that the turbines and boilers suffer from a loss of neatness and simplicity due to the bulky piping. This applies even to the very pleasing two-casing tandem-turbines in the Columbia Park Station (Fig. 14). It is for this reason that the practice has recently arisen of reheating by means of highly superheated live steam. All the steam before entering the turbine can be led through heaters where it warms up the expanded steam; or, most of the steam from the boilers can be led directly to the turbine and a small part directed to the reheaters where it is condensed. This last method seems particularly to favour the application of reheating. Resuperheating by live steam and also by condensing part of it can be employed together. Of the two German Stations which now use reheating, one (Mannheim Power Station with 1400 lb./sq. in. (100 kg./cm.²)

pressure) (14) uses this combined system, in the other part of the live steam is condensed (Ilse-Renate Lignite Mine industrial power station with 1400 lb./sq. in. (100 kg./cm.²) steam pressure (15).

Recently, still another method has been investigated and will later be applied in the American industrial station of the *Dow Chemical Co.*. With this arrangement the high condensing temperature and specific heat of diphenyloxide at elevated temperatures will be employed. On these points the substance resembles mercury without, however, being so dangerous or so expensive. The diphenyloxide will be heated by the flue gases to the required temperature and led in the liquid state to the reheaters in the engine-room. The heat exchange will be better than with a steam-heater arrangement, and the large pipes to the boilers and back will be avoided. The practical results of this method are awaited with interest.

The best point to extract the steam from the turbine for reheating, and also the most suitable reheat temperature, require a thorough investigation in each individual case.

The practical value of reheating is usually a matter of opinion. In all cases the plant is much complicated, more stages are needed in the turbine for the greater heat drop, better materials are required for the higher temperatures and, whether the reheating is done in the boiler house or in separate heaters, the operation of the station is more delicate and requires greater supervision. For these reasons chiefly, the application of reheating, in Europe especially, is making only a slow progress.

d. Regenerative feed-water heating

Regenerative feed-water heating is the extraction of steam at different points of the turbine to heat the boiler feed by stages. It was already employed occasionally, with reciprocating steam engines many years ago. Its systematic development is, however, mainly due to the modern steam turbine. Steam having transformed part of its internal energy into work, is bled from an intermediate stage and used for preheating the condensate of the turbine before it is returned to the boilers. By this means, the latent heat of the bled steam is not lost in the cooling water of the condenser but remains in the cycle. If steam is only bled at one point, the pressure at that point will determine approximately the feed-heating temperature. With multi-stage feed-heating the condensate is first heated by low-pressure steam up to an intermediate temperature and then by steam at higher pressures up to the final feed-heating temperature. Naturally, the greatest thermal gain is obtained when the number of stages is great or, mathematically expressed, infinite.

The top group of curves in Fig. 15 shows the improvement in thermal efficiency in function of the feed-heating temperature for various initial pressures and temperatures, and for one or two-stage feed-heating. It can be seen that the process is more economical the higher the initial steam pressure. This is to be expected, since, for high initial pressures, the bled steam uses a greater heat drop and does more work before leaving the turbine. Another obvious fact that these curves show is that, all things being equal, two-stage feed-heating is more economical than single-stage.

The lower group of curves gives the gain in thermal efficiency in function of the initial pressure for feed-heating with one, two or an infinite number of stages, and gives the most favourable feed-heating temperature in each case. For instance, with 1150 lb./sq. in. (80 kg./cm.²) gauge initial pressure and an infinite number of preheating stages, 16.5% improvement in thermal efficiency

(14) See p. 181 and *F. Marguerre:* Z. VdI. 73 (1929) pp. 913 and 993.
(15) See p. 180 and *O. Schöne:* Proceedings of the 4th meeting of the Commission on H.P. plants of the "Vereinigung der Elektrizitätswerke" (Germany) (1929) p. 71.

could be obtained, the feed temperature which corresponds to this best case being 560° F. (294° C.), or the saturation temperature at 1150 lb./sq. in. (80 kg./cm.^{2}). With two-stage heating the gain is theoretically at the most 11.5%, the feed temperature being 383° F. (195° C.). Naturally, the number of bleeder branches is limited for practical reasons.

When next investigating how much differences in the live steam and condensing conditions affect the gain obtainable by feed-heating, it is seen that, for a given number of heaters, the initial pressure and temperature have only little influence. Changes in vacuum only slightly affect the gain, which usually increases with the vacuum. The most important factor is the number of stages; when, for several live steam pressures, the best feed-heating temperature is plotted against the number of stages, it is seen that up to four stages the rise is fairly rapid, whilst after that it is less pronounced. The higher the initial pressure, the higher, naturally, is the feed temperature.

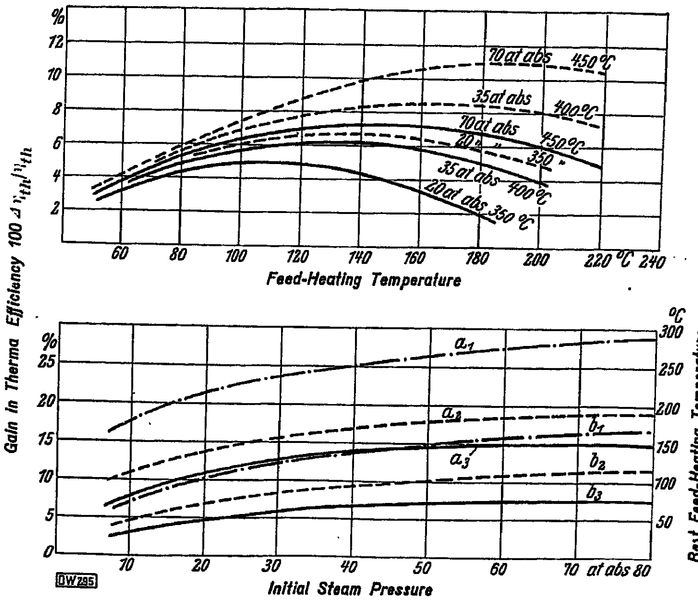


Fig. 15. Gain in thermal efficiency of steam turbines due to regenerative feed-water heating

———— Single-stage bleeding
- - - - - - Two-stage bleeding
—·—·—· Infinite-stage bleeding
a_1, a_2, a_3 = Best feed-heating temperature (righthand scale)
b_1, b_2, b_3 = Gain in thermal efficiency (lefthand scale)

The pressures at the bleeder branches have also an influence. As an approximation, it can be said that the best point to bleed for single-stage feed-heating will be near the middle of the heat drop; whilst for multi-stage feed-heating the heat drop should be divided into a corresponding number of equal portions. The bled steam quantities will vary between about 15% for single-stage and 30% for three or four-stage feed-heating.

Apart from these theoretical thermal gains, bleeding the main turbine has the additional advantage that, for a constant output, the load of the H.P. part is increased at the expense of that of the L.P. part. Thus the amount of steam passing through the H.P. part is increased, consequently, the steam passages and blade lengths are enlarged, and the H.P. efficiency improved. At the same time the flow through the L.P. part and the leaving loss are diminished, improving the L.P. efficiency. If, on the other hand, a constant leaving loss is taken, the limit capacity of the turbine is increased because, for a given speed, it is determined chiefly by the maximum diameter and blade length of the last stage.

Feed-heating has undoubtedly the advantage that a portion of the water which is carried along by the steam, or is condensed, is extracted at the bleeder branches. In order to do this an ample gap must be left in the blading so that the mixture of steam and water looses its speed and the moisture then separates.

The leaving energy is, therefore, lost. The erosion of the blades due to condensed water may very probably be diminished by suitably arranged bleeder branches. It should be remembered, however, that a large part of the condensation takes place at the lowest pressures, consequently, only after the last bleeding point.

The adoption of feed-heating has been very rapid. In Europe it is now the custom to provide nearly all ordinary condensing turbines with one or two branches; a greater number of heaters is not considered economical. In America, the number is increased, especially for large machines up to five stages. *Westinghouse,* for instance, recommend two-stage feed-heating for machines up to 10,000 kw., three-stage up to 15,000 kw. and for all larger machines four or five-stage heating.

Instead of heating the feed-water by means of bled steam from the main turbine, a separate turbine can be provided in the station for feed-heating purposes. It may be designed as a simple back-pressure machine, or it may have one or more bleeder branches. It can be used as the house-set. Economically the previous arrangement is somewhat superior, as the main turbine will have a slightly better efficiency than the small feed-heating unit, both in the H.P. part, with its large blade areas and in the L.P. part, with the reduced leaving loss. On the other hand, if feed-heating is to be arranged from the main turbine without too great pressure and temperature losses through long pipe lines, the heaters must be placed near the main condenser, and there is an accumulation of pipes, valves and vessels. For this reason separate feed-heating is sometimes advisable in large plants; it will have plenty of space to be laid out, it will be completely separate from the main turbine and its condensers, and will be more accessible. Another advantage is that there will then be a small set always ready to be started up, it can be coupled to the auxiliary current supply and it will be easier to get the station working than if the main sets have to be started up by themselves. Still another advantage may be noted from the fact that, with a separate feed-heating unit, the bled steam and feed temperatures are independent of the load on the main turbine. They can be regulated more easily than when they vary with the output, as they do in the first system. The super-power station of Klingenberg, near Berlin, is an example of a plant of this kind (16).

The flue gases used formerly to be generally employed for heating the feed-water in the economiser after leaving the boilers and superheaters. It is hardly necessary to say that, if a regenerative cycle is to be economical, these gases should be used in another way, such as in an air heater.

If reheating and feed-heating are employed together in a turbine, the gains are practically independent of each other, and the total improvement is equal to the sum of the individual gains.

e. Power generation from process steam

In a condensing turbine the total heat of the exhaust steam is absorbed by the cooling water and is lost. However high, therefore, the thermodynamic efficiency of a turbine may be, its thermal efficiency will always be very low. Neither will there be any noteworthy improvement if reheating or feed-heating, as described above, are used, or even if the two systems are used together. The gain will only be a small percentage. However much pains and ingenuity may be expended in improving condensing steam-engines, whether they be of the reciprocating or turbine type, and whatever high degree of perfection is reached, they are and will remain inherently uneconomic power generators.

(16) See Z. VdI. 71 (1927) pp. 1829/1912, especially the papers of *H. Denecke* (p. 1877), *E. A. Kraft* (p. 1869) and *R. Tröger* (p. 1902).

These conditions are changed if, apart from the mechanical or electrical energy, heat is also required; it may be possible to supply this by the steam at a low or medium pressure. In such a combined power and heat producing station (17), the whole of the steam required for the process, or utilizable in the turbine, is generated at a considerably higher pressure than is required for heating, and does work when expanding in the turbine. The temperature required for the process steam determines the back-pressure, as, usually, only saturated or slightly superheated steam is required. The pressure and temperature drops in the piping should also be taken into account. We shall not repeat the well-known comparison between the heat balance of a condensing and a back-pressure turbine. It will, however, be mentioned that if there is a demand for the power that can be extracted from the process steam and if the heat contained in this steam is all required, a plant of this kind will have a very high efficiency as it has only small mechanical, electrical and radiation losses; even for low powers it can often compete successfully with the largest steam and hydraulic power stations.

The efficiency usually goes down if the demand for steam is greater than the turbine exhaust, and the excess has to be made up by throttling live steam. If more power is required than can be obtained from the process steam, an extraction condensing turbine will have to be used instead of a back-pressure machine, and the efficiency will generally decrease; the decrease will be especially great if the conditions vary greatly with the load. Nevertheless, these turbines (back-pressure, single or multiple-extraction and extraction-back-pressure) are one of the most important means of increasing the economy of steam power plants. Their characteristic is that they can supply both the power and the heat required in industrial concerns, and for this reason they are sometimes called industrial turbines, to distinguish them from the condensing turbines of large power stations. A very careful calculation of the commercial value of the station should be made before it is laid out. The turbine itself should be designed correctly, and the governing, which may be very complicated, should be correct both from a mechanical and a thermodynamic point of view.

A few words will be said later about the commercial value of industrial power plants. The question is very important and varies with each individual case. Before we deal with it, reference must be made to a particular arrangement of combined power and heat generating plant.

The principle which underlies the combination of power and heat production can only then be fully effective when the narrow limits of requirements of an individual industrial concern, namely well defined demand for energy on the one side and for heating steam on the other, are abandonned and a larger number of individual concerns are linked up together to a simple power station. Of the great number of possible combinations of this kind only two will be mentioned. Firstly, the factory generates, by means of a back-pressure turbine, as much power as it can from the process steam it requires; any surplus current is sent away, or any deficiency is made good by the main power station, and thus, according to the daily or seasonal demand, current may be either supplied or received. For this mode of operation, power may sometimes be only a by-product. Secondly, the power station may generate part of its current by a back-pressure turbine, or, if need be, by an extraction back-pressure turbine, and is able to deliver to its clients not only power, but also process steam, which can be, if necessary, at different pressures. In this case steam is a by-product for the power station, it gives out waste heat. The first solution requires small back-pressure turbines in a great

(17) See *H. Treitel:* "German practice in exhaust steam engineering", p. 1153, "The Transactions of the first World Power Conference", Vol. 2 (London: Percy Lund Humphries & Co. Ltd. 1924).

number of plants, and a large number of transmission lines to the centre of distribution. The second solution requires a small number of powerful back-pressure turbines in a large station and a widespread net of steam piping. Both systems of linking have been employed for the last few years; the first method is tending to become from year to year more frequent, as large, complicated and expensive pipe lines are not at present popular.

Concerning the design of industrial turbines, the first data can be obtained from the steam charts, as was the case for condensing turbines. The factory usually requires a determined amount of heat and steam, the back-pressure is, therefore, generally given and in most cases this is the starting point for the calculations of industrial turbines. It has already been shown how important it is to lower the back-pressure as much as possible. Sometimes this can be done by shortening or improving the steam piping. If, for instance, the back-pressure can be lowered by simple means from 28 to 14 lb./sq. in. (2 to 1 kg./cm.²) absolute, then, as seen from Fig. 8, the theoretical gain in power will be the same as when the initial pressure is increased from 285 to 1000 lb./sq. in. (20 to 70 kg./cm.²) absolute. Once the back-pressure has been fixed, the live steam conditions can be chosen so as to obtain, within certain limits, the desired power from the process steam (Fig. 16). If for instance, 143 million B.Th.U. (36 million kcal.) per hour are required from steam at 300° F. (150° C.) or the saturation temperature at 71 lb./sq. in. (5 kg./cm.²) absolute (I = 1180 B.Th.U./lb. = 656 kcal./kg.) then, if the boilers and machines are assumed to be perfect, about 55 tons per hour of steam will have to be generated. The factory may require 1700 kw. in electrical power; each ton per hour of steam must therefore produce 31 kw. and this can be done with the very moderate initial conditions of about 120 lb./sq. in. (8.5 kg./cm.²) absolute and 445° F. (230° C.) T.T.. If the temperature is raised as high as the present day materials allow, or to about 930° F. (500° C.) and the pressure is taken as 925 lb./sq. in. (65 kg./cm.²) absolute, 182 kw. could be produced for every ton per hour of steam. Therefore 55 × 182, or 10,000 kw. could be generated and 8000 kw. surplus power could be sold. Thus, by increasing the heat consumption by about 20%, the gain in output is something like 500% for this example.

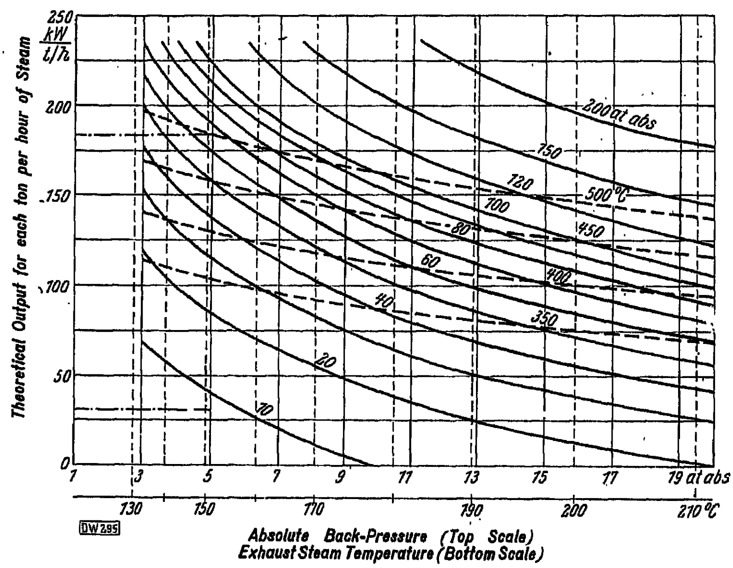


Fig. 16. Output theoretically attainable by an expansion down to saturation point

——— Initial steam pressure
- - - - Initial steam temperature

The fact that these values are based on an ideal machine does not alter the fundamental importance of the point. For actual machines, if the initial pressure and temperature are raised by a small amount, the thermal efficiency

for the production of electrical power by a back-pressure machine will remain in the neighbourhood of 100%. The conditions chosen from the steam tables may be feasible from an engineering point of view, only a comparison of the costs, however, can decide if they are economically suitable. A gain in thermal efficiency by increasing the pressure causes a rise in the capital, running and maintenance charges (18).

To quote a characteristic example let it be supposed that a factory requires about 26 tons of steam per hour at 300° F. (150° C.) and 850 kw. in electrical power. The factory may be a sugar factory and will work each year for a season of three months. It will now be compared with a second factory with the same boiler and power requirements, but which works the whole year through with 70% load factor. With 140 lb./sq. in. (10 kg./cm.²) absolute steam pressure, the internal demand for power can be completely met, any higher pressure will give an excess of power which may be sold. For the sugar

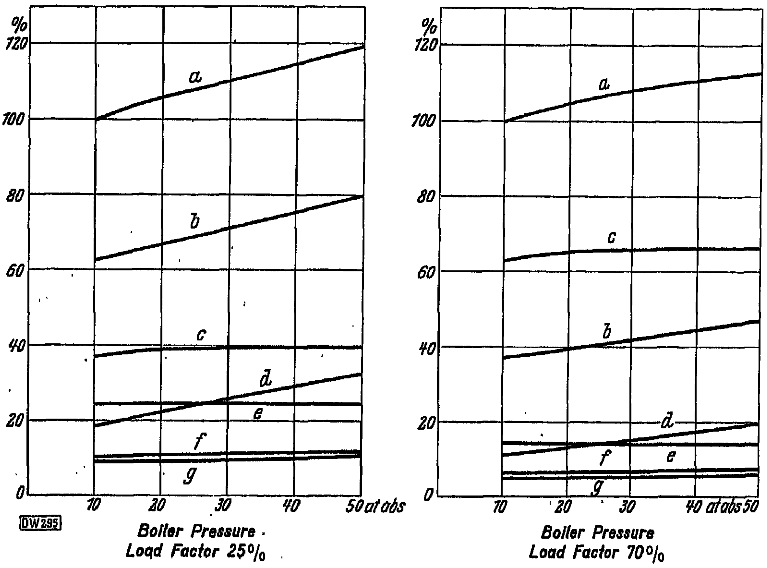


Fig. 17. Comparison of the operating costs of two back-pressure turbine plants each for 26 tons per hour of steam in relation to the boiler pressure

a = Total cost
b = Fixed charges
c = Fuel cost
d = Boiler, e = Building, f = Equipment, g = Turbine — Interest and depreciation

factory, the annual costs, for a present day average rate of interest, will be distributed between the capital and the fuel charge as shown on the left of Fig. 17. Supervision and the other operating charges have been neglected as only a comparison is intended. The fixed charges are the most important, therefore, when the boiler pressure is increased and the fixed charges rise steeply, the total cost will vary in a similar way. If the yearly running time is increased the situation changes. The fuel charges become more important than the capital charges; as shown on the right of Fig. 17, a rise in the boiler pressure increases the total cost relatively less.

On the left of Fig. 18 is given the power obtainable for varying boiler pressures. The surplus power is shown shaded, and is to be regarded as a saleable product. On the right of Fig. 18 is shown the price the factory has

(18) See *E. Reutlinger-M. Gerbel:* "Kraft- und Wärmewirtschaft in der Industrie" (Berlin: J. Springer 1927).

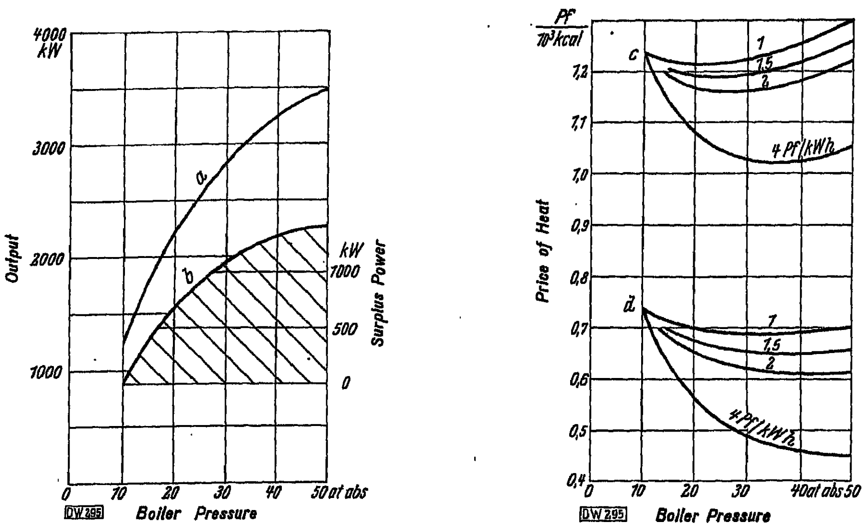


Fig. 18. Surplus power and price of heat of two back-pressure turbine plants for various prices of current in relation to the boiler pressure

a = Theoretical output c = Load factor 25%
b = kw. at terminals d = Load factor 70%

to pay for every 4000 B.Th.U. (1000 kcal.) from steam, or for the equivalent electrical energy. The curves are made out for several prices of power, and the revenue from the sale of surplus current has been subtracted from the total cost. For every saleable price of power there is, therefore, a corresponding economical boiler pressure. If the sugar factory can sell current at 4 Pfennigs or if the plant with 70% load factor can sell it at 2 Pfennigs, the best pressure will be less than 710 lb./sq. in. (50 kg./cm.²) absolute. If the factory with 70% load factor can sell current at 4 Pfennigs this pressure would be greater than 710 lb./sq. in. (50 kg./cm.²). For 1.5 Pfennigs, for instance, the best pressure would be about 355 lb./sq. in. (25 kg./cm.²) absolute for the sugar factory and there would be about 1800 kw. generated of which 950 kw. would be surplus power. For the other plant and the same price for current a boiler pressure of as much as 540 lb./sq. in. (38 kg./cm.²) absolute could be taken and a total power of 2150 kw. or 1300 kw. excess power could be obtained. In this way the best boiler pressure can be determined for an industrial station. As the state of the steam at the exhaust is usually given, the initial temperature can also be found. It is incorrect, therefore, to affirm, as is now often done, that economical modern machines must always require higher and higher pressures. It is much better to determine the right pressure by a careful calculation which takes into account the particular conditions at the station. In this way often medium, and sometimes low pressures are arrived at.

Finally, when all the external conditions such as the quantities, pressures and temperatures of steam, the loads, and also, what is very important, the average values (i. e. the operation conditions which will occur most frequently) are fixed, the problem is to find the best turbine for the conditions. Examples of industrial turbines constructed for all sizes and powers are illustrated on pages 131 to 149 in detail.

The advantages and possibilities of the application in general of industrial steam turbines, which has only been briefly discussed here, and in particular, the coupling of back-pressure machines to large power stations, can hardly be overestimated. They are, to a large extent, due to the great progress

in the construction of modern steam turbines which enables high pressures and temperatures to be used with a good efficiency in even medium or small sized back-pressure turbines. It seems as if they were destined to play an important part in the near future in the systematic economy of energy and fuel. Industrial stations are, perhaps, even more justified in using very high pressure steam than ordinary power stations; they are evolving in this direction, as is proved by the case of the 24,000 kw. plant with two 1400 lb./sq. in. (100 kg./cm.^{2}) turbines, in a large lignite briquette factory in Central Germany. These turbines will be fully described later.

2. Design

a. Steam path

The working process of steam in the blades of a turbine, which can be essentially considered as a flow of a compressible fluid through passages of varying section, is accompanied by losses. These may be classified into individual losses, the nature of which is known even if their true value is not. Only four will be named here: surface friction, shock loss, loss due to bend and the bridging over of clearance loss (19). In the theory of steam turbines these individual losses are usually all considered together, they are taken account of in calculations as the losses due to the flow by means of the so-called velocity coefficients φ for the stationary blades and ψ for the moving blades. The flow losses in steam turbines depend on many physical and geometrical factors, the most important of which are the steam velocity, the shape and size of the steam passages and their boundary surfaces, the blades and clearances, also the pitch, length and surface finish of the blades; finally, the diameter of the stage and the conditions of the steam also have an effect. Of these quantities, the steam velocity is the most important, or at least it has been so considered for a long time. The absolute velocity should, naturally, be taken for a fixed blade, and the relative velocity for a moving blade. For this reason steam turbine engineers concentrated their efforts on obtaining, by laboratory research and from tests on existing turbines, reliable data on the relation between the flow losses and the steam velocity. The question was, therefore, to discover if the flow losses are less at high or at low velocities, or if, perhaps, there is a minimum at a medium speed. The answer would be found when the velocity coefficients φ and ψ are plotted as a function of the steam velocity. However old this question may be, and whatever the pains that have been taken in seeking its solution, opinions on the subject are always greatly different, and disputes are still as numerous as ever.

From tests which *Josse* (20) and *Christlein* (21) made nearly 20 years ago, it appears that the flow losses through guide vanes with parallel walls decrease as the steam velocity increases, and the best velocity for fixed blades lies above the critical velocity (i. e. velocity of sound) and is about 2000 ft./sec. (600 m./sec.); for moving blades it would be in the neighbourhood of the critical velocity or about 1500 ft./sec. (450 m./sec.). Fig. 19 shows the results of the above tests. It will be seen that φ is higher for greater steam velocities than it is for lower ones, and that ψ increases to about the critical velocity and then decreases. High steam velocities mean large heat drops in each stage and the number of stages will be small. If a good thermodynamic efficiency is required the ratio of the peripheral to the steam velocity may not vary much;

(19) G. *Flügel* in his dissertation "Über die näherungsweise Erfassung der Strömungsverluste und das Krümmerproblem" (In "Hydraulische Probleme", Berlin: VdI.-Verlag 1926, p. 133) gives a general classification of flow losses.

(20) Jahrbuch der Schiffbautechnischen Gesellschaft 13 (1912) p. 340.

(21) Z. VdI. 55 (1911) p. 2081.

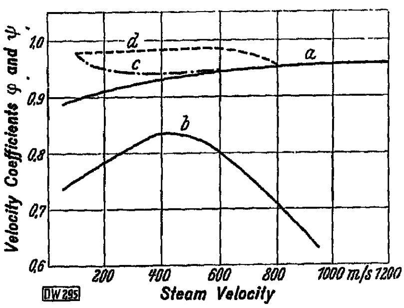


Fig. 19. Velocity coefficients φ and ψ for steam turbine blades, stationary and rotating

$a = \varphi$, $b = \psi$ } From tests of *Josse* and *Christlein*
$c = \varphi$ From tests of the *Steam-Nozzles Research Committee*
$d = \varphi$ From American tests

therefore, high peripheral velocities and large diameters will be needed. The attainment of a better efficiency by reducing the number of stages and the capital costs was, naturally, very attractive. As a result of these tests and from a natural desire to produce a cheap machine, steam turbine design showed, until about 1920, a general tendency to fewer stages and larger diameters.

Since then tests, some of which were very costly, have been made in many different places, such as those by the *A.E.G.* (22), by *B.B.C.* (22) and by *Stodola* (22). The *British Steam-Nozzles Research Committee* has carried out researches since 1914 on a very extensive plan. Their tests by the impact method have shown (23) for both nozzles with parallel walls (with parallel guiding of the leaving steam) and with the so-called closed steam jet, exactly the opposite to the results of *Josse* and *Christlein*. The losses decrease generally with the steam velocity and for some tests it is remarkable that a minimum is reached for velocities between 260 and 330 ft./sec. (80 and 100 m./sec.). These results are also in disagreement with the results from some recent American tests (24) which use the reaction method. An average curve of these and of the English results is given in Fig. 19. No conclusive proof of the correctness of either one or the other of these curves has been produced as such tests are influenced by an enormous number of factors and require great skill and accuracy. It is difficult to say whether a research based on the impulse method, or one based on the reaction method has more sources of error and this question cannot be discussed here. Tests of this kind may be very important for a scientific knowledge of steam flow and, to a certain extent, for the design of turbines when properly applied. It must, however, be emphatically and clearly stated that they do not give any general or precise information on the steam velocity for obtaining the best thermodynamic efficiency for any particular turbine type or stage. In other words, if all the stages of a turbine were designed for steam velocities which had the highest values for φ and ψ, the turbine would by no means give the best thermodynamic efficiency. The velocity should rather be chosen so that the other factors which influence the blade efficiency, some of which depend also on the steam velocity, have the most favourable values. Tests on actual turbine stages, with stationary nozzles and rotating blades, have not as yet been published, neither have they probably been made with the intention of finding the velocity coefficient. The proper inducement is missing, at least as far as turbine builders are concerned, as it has now been found that other factors of a secondary nature have a much greater influence on the thermodynamic efficiency; this subject will be further discussed later.

Another opinion has been gradually spreading concerning the losses of the flow through any passage of varying section and, in particular, through a turbine stage consisting of nozzles and blades. According to this opinion the losses would be caused by the so-called compression shock, and they would in-

(22) *A. Stodola* "Dampf- und Gasturbinen" 6th (German) Edition pp. 121/127 (Berlin: J. Springer 1924); "Steam and Gas Turbines" pp. 119/124 (New York: Mc. Graw & Hill 1927).

(23) Up to the present six reports have been published. See Engineering 115 (1923) pp. 356, 377, 380, 501; 117 (1924) p. 681; 119 (1925) pp. 614 & 617; 125 (1928) pp. 107, 109, 117; 129 (1930) pp. 296, 319 & 361.

(24) See *G. B. Warren* and *J. H. Keenan:* "A machine for testing steam nozzles by the reaction method", Mech. Eng. 48 (1926) p. 227.

crease, for a given steam velocity, with the pressure and, for a given pressure, with the steam velocity. This would mean for the steam turbine theory that the velocity coefficients φ and ψ would not only depend on the steam velocity (and perhaps other factors as well), but also on the pressure or density of the steam. Consequently, it would seem that if a given turbine were to work over different pressure ranges, with the same heat drop per stage and the same steam and blade velocities, the efficiency would be worse for the higher pressures than for the lower ones. As *Stodola* (25) also points out, most of the resistance through the blades should be due, if secondary currents are ignored, to the compression and re-expansion of the steam in the bend. In spite of this, no test results are known which would prove the dependence of the flow losses upon the pressure or density of the steam, at least there are no results which would not allow of another interpretation. The theory may not be true for the general case, and it has not been proved correct for the present day range of pressures, but it certainly has a correct nucleus. Under certain conditions, as in the case of high pressures, it gives good design, by which is meant turbines with high thermodynamic efficiencies. In this way *Lösel* succeeded in developing the multi-stage back-pressure turbine of the *Erste Brünner* type which has now been so generally adopted.

Let us consider the case of a turbine with full admission which, especially for machines of small capacity, should be used if possible when a high efficiency is required. Low steam velocities need ample steam passages, consequently, long blades. The larger the ratio of the useful area to the leakage area, the smaller is the leakage loss for the stage. Therefore, the adoption of low steam velocities, whether the turbine be impulse or reaction, influences the efficiency favourably. The raising of the steam quantity or the load would have the same result as lowering the steam velocity. This leads to the conclusion that, as far as the leakage loss is concerned, it is always advisable to choose the smallest possible steam velocities. They should be smaller, the greater the pressure in turbine, stage or group of stages. For this reason the steam velocity should be chosen, for a given steam volume, so that the leakage loss does not surpass what is considered a permissible limit. Hence, it would be wrong to choose in advance a certain range of velocities. For a given initial pressure the steam quantities and volume vary with the output; therefore, the velocity could be chosen all the higher, and the number of stages all the lower, as the output is increased.

When the influence of the blade length is considered, similar conclusions are reached. In reaction turbines the clearances which determine the leakage loss depend directly on the blading. For reasons of safety the clearance must not be less than a certain fraction of the diameter. Hence, for small blade heights large diameters cannot be chosen as the leakage loss would be too great. In impulse turbines, on the other hand, the leakage usually takes place across the diaphragm packing glands, it can thus be considered, to some extent, as constant, or at least as not depending on the blade diameter. Therefore, if only the leakage loss, or the ratio of the useful area to the leakage area, is considered, it would make no difference to the impulse turbine whether the steam passage is formed by long blades with small diameters or short blades with large diameters. For small blade lengths, the increase in height between the nozzles and the blades, or the overlap, must also be small. It would be more difficult to manufacture this dimension with sufficient accuracy when the diameter is large than when it is small. Moreover, with small diameters and long blades the overlap may be chosen larger.

Observations and test results have proved beyond doubt, that not only does the efficiency depend on the useful area, but also on the height of the steam jet;

(25) 6th (German) Edition p. 153; English Edition p. 156.

in other words, the radial length of nozzles and blades has a decided influence on the stage efficiency. With decreasing blade length the blade efficiency falls rapidly and with very small blade lengths it reaches extraordinarily low values.

This decrease may, at first sight, be accounted for exclusively by friction on the walls, such as occurs in steam pipes. According to the usual formulae the losses become larger as the ratio of the perimeter of the section to the area increases. For blades of small length this ratio is very large and is, therefore, unfavourable. However, if the loss from friction against the walls is calculated in this way, it will be found that the bad blade efficiencies met in practice will only be obtained when a much worse coefficient is taken than usually is found for the loss through ordinary steam passages. It must be concluded, therefore, that the flow through these small channels is accompanied by some peculiar additional losses. It may be that the regions of disturbed flow in the neighbourhood of the walls, the so-called *boundary flows*, are so near together that they hinder also each other; it may be that the steam entrained by virtue of injection action has a specially unfavourable influence on the narrow ring-like steam stream, or it may be some other disturbance. Whatever may be the cause, the result is beyond doubt, and has been established by many observations. Fig. 20, for instance, gives the results of some recent American tests on the subject. It will be seen that the efficiency of a nozzle of say 4 in. (100 mm.) radial height is about 97.5%, and for one of $^3/_{16}$ in. (5 mm.) it is only about 89%. The total decrease in the internal efficiency is, however, much greater. The lower curve shows this total efficiency, it takes account of the leakage loss, the boundary loss and the additional flow losses. Thus, if the internal efficiency for a stage is 85.2% for a nozzle 4 in. (100 mm.) high, it will fall to about 50% if the nozzle is reduced to $^3/_{16}$ in. (5 mm.). It may be mentioned that *Stodola* (26) has already pointed out that the efficiency and velocity coefficients both decrease with reduced blade length. He even gives a curve which agrees in general with the results of other investigators, but still gives too high values for the velocity coefficients.

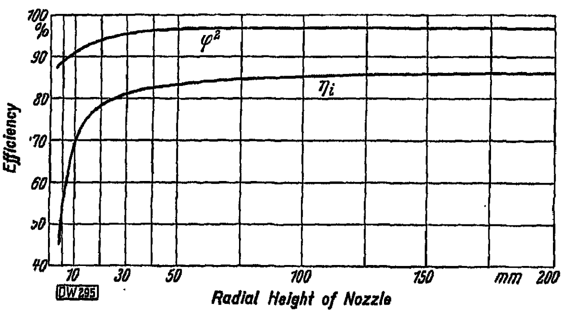


Fig. 20. Efficiency φ^2 of a nozzle and efficiency η_i of an impulse stage dependent on the radial width of the steam path (height of nozzle)

A curve as given in Fig. 20 could be used directly for turbine calculations. A similar curve is also obtained when the efficiency of multi-stage turbines is plotted as a function of the average steam volume flowing through, the turbines being built for different values of the volume, consequently different lengths of the blades, but otherwise alike. This curve is shown in Fig. 21 for back-pressure turbines of the impulse type and with a large number of discs of equal diameter. It is the average curve of a large number of test results. The ordinate is the internal efficiency of the turbine; it includes all the losses of the turbine, except the mechanical loss. This curve can, naturally, only be established for machines with a small heat drop, such as back-pressure turbines. For other machines, it would not be permissible to take the mean volume in this way and the condition of constant diameters could not be realized. In

(26) 6th (German) Edition p. 151; English Edition p. 183.

addition, an expansion past the saturation point would be disturbed by the wetness of steam.

If an analytical equation is preferred to a graphical representation, the efficiency curve can be assimilated with sufficient accuracy to a hyperbola

$$\eta = \eta_{\infty} \cdot \left(1 - \frac{C}{l}\right)$$

where

η = mean blade efficiency,

η_{∞} = mean blade efficiency for infinitely long blades,

l = mean blade length in inches and

C = a constant, also in inches, which may be taken to be double the thickness of the disturbed layer in the neighbourhood of the walls.

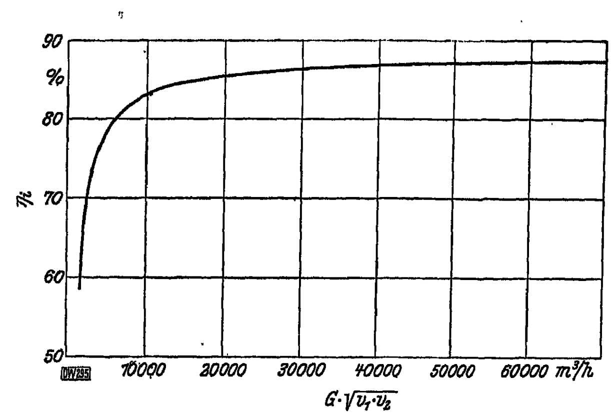


Fig. 21. Internal efficiency η_i of multi-stage, impulse, back-pressure turbines as a function of the mean steam volume, $G\sqrt{v_1 \cdot v_2}$ m.³/hour

G = Weight of steam, kg./hour
v_1 = Specific volume at the beginning of the expansion, m.³/kg.
v_2 = Specific volume at the end of an adiabatic expansion, m.³/kg.

In the case of a multi-stage turbine, with same mean diameter in every stage and where the heat drops per stage, the nozzle and blade angles, and the speed of rotation are given, the blade heights l are directly proportionate to the total volume. This is the quotient of the steam quantity G in lb./hour, an the specific weight γ in lb./cu. in.. As for superheated steam the specific weight is nearly directly proportionate to the absolute pressure p in lb./sq. in., we have

$$l = k \cdot \frac{G}{p},$$

where k is another constant (with dimensions of hour/in.). This relation may be extended to the whole turbine, taking for the pressure the geometrical average of the initial and final pressures,

$$p = \sqrt{p_1 \cdot p_2}$$

then,

$$l = k \cdot \frac{G}{\sqrt{p_1 \cdot p_2}}.$$

This value can be substituted in the first formula,

$$\eta = \eta_{\infty} \cdot \left(1 - \frac{C}{k} \cdot \frac{\sqrt{p_1 \cdot p_2}}{G}\right),$$

where $\frac{C}{k}$ is a constant depending on the units chosen; its dimensions are sq. in./hour. By means of these or similar formulae the efficiency of a multi-stage impulse turbine can be calculated on the basis of the original suppositions, or at least it can be estimated approximately. For a single-stage turbine the formula can be adapted to suit.

If η_{∞} for a single stage is taken about 87% and the constant C about 0.079 in. (2.0 mm.), the values of tests and calculations shown in the two diagrams will be in reasonable agreement with the values from the formula. In Fig. 21 η_{∞} is a trifle higher, or about 88%; this is due to the gain from the reheating effect. The constant $\frac{C}{k}$ has approximately the value 85.3 sq. in./h. (550 cm.²/h.). It must be clearly stated, however, that if values are given here they can only be taken as approximations. The chief object of the curves is to show that the

influence of the blade lengths on the internal efficiency follows closely a hyperbolical law. The stage efficiency, for instance, cannot be directly determined by test, it can only be deduced from the internal efficiency of a whole turbine after certain assumptions have been made. These limitations have always to be kept in mind when use is made of these curves and figures.

It may seem that too much space is devoted to the influence of the blade length, it should, however, be remembered that this influence has only been clearly recognized in recent years, when it has been endeavoured to utilize high pressure steam in an efficient manner in back-pressure and condensing turbines even of smaller and smallest capacities. Previously, the height of nozzles and blades for impulse turbines with partial or full admission were also not taken less than certain values; in this instance the limit was fixed mainly for manufacturing reasons, especially in the case of cast-in nozzles or guide vanes. With modern turbines, however, when a high economy is required, the first stage, on which the regulation is done, is usually the only one with partial admission. For small outputs and high pressures small blade heights are then obtained, and the efficiency is reduced. This decrease can be so marked that sometimes, for a given back-pressure and output, no improvement in specific steam consumption is obtained when the initial pressure is raised over a certain limit.

The losses due to the rotation of the rotor and blades in the steam are usually considered together under the two headings of disc friction and ventilation losses. They are only of importance for small outputs. Their sum appears to be smaller for a large number of small stages, such as are obtained with low steam velocities, than it is when the same heat drop is used in a small number of stages with large diameters.

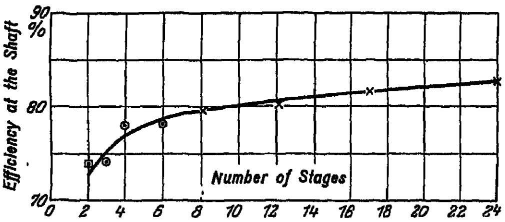


Fig. 22. *Amer. G.E.C.*, efficiencies of 4000 kw. turbines with various numbers of stages. Speed 3600 R.P.M.

⊡ = Two two-row stages
⊙ = Two-row first stage
× = Turbine with 8 to 24 single-row stages

The heat that can be regained by the reheat, is the increase of the heat drop in the turbine above that corresponding to the adiabatic, and is due to the friction losses in the steam being converted back into heat. This gain is larger for machines with many stages than for those with only a few. When the number of stages is large this increase is, however, only very small (27).

All these considerations lead to the conclusion that, when a high thermodynamic efficiency is the chief aim and other factors are only of secondary importance, machines with a large number of stages and small steam velocities are usually better than machines with fewer stages and greater steam velocities. When a good steam consumption is of first importance turbines are now made with a large number of stages; hence, two or more casings may be required. This is equally true for both impulse and reaction turbines. It will be seen that if the above principles are strictly applied, machines of small outputs would have to have a larger number of stages than those for large outputs. When the losses from the external glands and the bearings, and the cost of manufacture are considered, it will be found that the application of these rules is limited.

The recognition of the thermodynamic superiority of turbines with many stages is one of the oldest achievements of steam turbine design. It will be remembered that the first turbine producing a large output with sufficient reliability and economy, or *Parsons'* turbine, was multi-stage. Also, in May 1913,

(27) See *A. Weiwerka:* "Die rückgewinnbare Wärme bei Hochdruckdampfturbinen", Archiv f. Wärmewirtschaft 7 (1926) p. 189.

for instance, the chief engineer of the Lynn works of the *Amer. G.E.C., Rice,* estimated the relation between the efficiency of condensing turbines and the number of stages (Fig. 22). Although the values are only based on calculations, they have been verified in practice as being a good approximation. The curve is valid for machines of medium size and is meant to show that turbines with a moderate number of stages are only slightly less efficient than those with a very large number. In other words, for a turbine of good efficiency only a very small improvement can be obtained by a large increase in the number of stages and a great extra cost. This is the reason why turbines of average efficiency could meet the economic requirements as long as the price for fuel was moderate. Fig. 23 shows the section through a single-casing turbine of 40 stages for 3000 kw. at 3600 R.P.M., built by the *Amer. G.E.C.* in 1916. Its efficiency must have been very good; nevertheless, this type of design was soon abandoned, since it was too expensive and the high cost was not justified when the price of

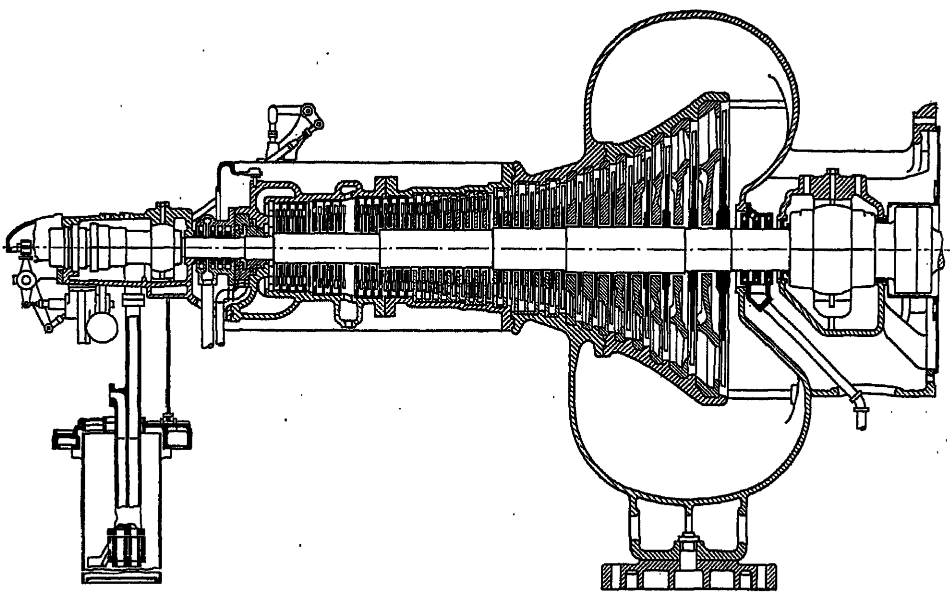


Fig. 23. *Amer. G.E.C.,* 3000 kw., 3600 R.P.M. high-pressure turbine

fuel decreased. Turbines of different efficiency may not be compared by considering only their weight and price. The purchaser should, in all cases, investigate the internal design of the turbine, even when the same steam consumptions are guaranteed, just as it used to be the custom to judge a reciprocating engine by the diameter of the cylinder, the stroke, the admission and the speed.

An increase in the speed of rotation has a similar effect to a decrease in the steam velocity. For instance, two impulse turbines may be compared, which work under the same conditions, have the same number of stages, the same heat drop per stage, the same nozzle angles, the same steam velocities and the same quality figures. Each turbine will have a constant stage diameter d_1 and d_2 respectively, the first one will run at n_1 R.P.M. and the second at n_2 R.P.M.. Then the diameter of the second turbine must be

$$d_2 = d_1 \cdot \frac{n_1}{n_2} \text{ in..}$$

The diameter is inversely proportionate to the speed. As the steam passages

in both turbines must be equal, the blade height of the second turbine must be

$$l_2 = l_1 \cdot \frac{n_2}{n_1} \text{ in.},$$

or the blade length varies in direct proportion to the speed of rotation. The favourable effect of higher speeds of rotation arises, therefore, from the increase in the blade length. The leakage losses, at least in the case of impulse turbines, will not be diminished as the ratio of the useful area to the leakage area will not have been altered. As a rigid shaft is usually required, and rightly, a higher speed may sometimes mean larger external and internal glands. The gain from the increase in blade height usually outweighs the constant or increased leakage losses. The correct speed should be determined by comparative calculations and designs.

One of the most important requirements, which must be considered with the losses seen from the velocity diagrams, is that the increase in blade area and length must always be regular. Losses due to eddies in dead spaces are thus avoided, and the profile of the moving and fixed blades, the shrouding and radial clearances should follow the steam flow, and make a regular shaped curve. In the H.P. zone, this requirement, especially when there is a large total volume of steam, can usually be easily satisfied as the specific volume increases only slowly. The case is different in the L.P. region. Here the volume increases rapidly, especially below atmospheric pressure, and a rapid increase in the blade areas and diameters is required. It will be difficult to design the steam passages as a regular and unbroken curve. The axial and radial clearances, and the overlapping of the elements have to be considered; secondary flows towards the tips or roots of the blades, as well as large radial components of the steam velocities, have to be avoided. This is one of the reasons why, if particularly good results have been obtained from superheated steam, of slowly changing volume, working in the H.P. part of a large turbine, it is quite wrong to draw direct conclusions about the efficiency of the L.P. part where wet steam is working under vacuum.

When designing the blading of a turbine for a given steam volume, the diameter of the first stage is determined by the smallest blade height which is considered admissible. The diameter of the last stage is fixed by the leaving loss together with the stress in the last wheel or drum and in the moving blades. The problem is now to design a regular shaped steam passage, without sudden changes in volume or blade length, without any dead spaces, and without using any angles which are too small to be manufactured. Examples of the way in which this problem can be solved will be seen later in sectional drawings through turbines; it is especially clear in Fig. 145 for a single-casing impulse turbine, and in Fig. 156 in the case of a two-casing combined impulse and reaction turbine.

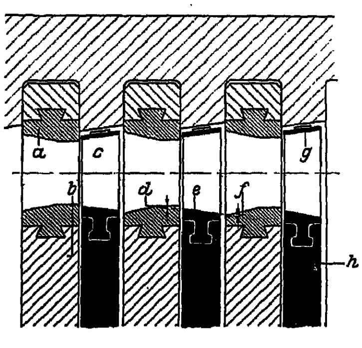


Fig. 24. *A.E.G.*, longitudinal section through the blading of a multi-stage impulse turbine

a = Nozzle
b = Diaphragm
c = Moving blade
d = Overlap at entrance
e = Distance piece
f = Overlap at Outlet
g = Coverband
h = Wheel

One way of obtaining a regular expansion for either impulse or reaction turbines is to use conical shrouding and caulking pieces. Fig. 24 shows such an arrangement for the H.P. part of a multi-stage impulse turbine. By means of a slanting coverband a too large difference in height between the nozzles and blades is avoided and there is no dead space or sucking of steam through the blades. Fig. 25 shows *Humboldt*'s design of the steam path in an impulse turbine; on the left is the

case of a turbine which is assembled axially, where the sealing edges may overlap; and on the right is the case of a horizontally split casing. In the latter case, after being inserted the rotor is pulled in an axial direction to the correct position by means of an adjusting device.

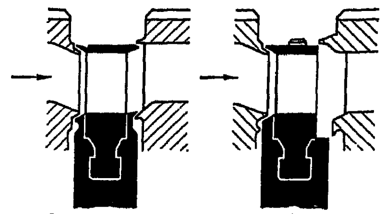


Fig. 25. *Humboldt*, sealing of the clearance of an impulse turbine for axial (a) and ordinary (b) methods of assembly

When considering the steam flow, naturally not only an axial section must be examined as this view cannot give a complete idea of the curved blades and varying sections. The steam path and areas in the direction of the flow should be borne in mind.

In reaction turbines, the only sealing against the inner surface of the casing and the outer surface of the drum, or any other internal wall of the steam passage, is often formed by the sharpened tips of the blades themselves without any shrouding. Then, as can be seen in the H.P. and I.P. parts in Fig. 143, a conical shaped casing or drum will help in producing a smooth flow. The ideal would, naturally, be a symmetrical form for the steam passage and the flow lines. In Fig. 46 this is achieved in the L.P. part, at the expense, however, of too short a blade in the first two stages and, consequently, with a too large leakage loss.

b. Quality figure

As is well known, the efficiency of a turbine stage depends, all things being equal, upon the value of u/c_0, which is the ratio of the blade velocity to the steam velocity corresponding to the adiabatic heat drop in the stage. The maximum value of the efficiency is for a value of u/c_0 of about 0.44 to 0.50 for pure impulse stages and of about 0.58 to 0.72 for stages with 50% reaction. However, the value of u/c_0 does not directly classify a stage or a complete turbine, as other quantities and losses have always to be considered. If the turbine has only a single stage it should be endeavoured to choose the speed so that the sum of the disc friction, ventilation and mechanical losses have the smallest possible value. Usually, high speeds up to 10,000 R.P.M. and more are required to satisfy this condition, and gearing is needed between the turbine and the machinery to be driven. The loss in the gears absorbs a certain amount of work which, like the disc friction and ventilation loss, is a constant and is approximately independent of the load. This loss may have its importance, especially at small loads and units, and the choice of the diameter and speed of single-stage turbines must be chosen with care in all cases.

In order to classify a multi-stage turbine in a similar manner, the ratio between a fictitious blade velocity $\sqrt{\Sigma u^2}$ ft./sec. to a fictitious steam velocity $c_0 = 223.8 \sqrt{h_0}$ ft./sec. ($c_0 = 91.5 \sqrt{h_0}$ m./sec.) could be used. h_0 would be the total adiabatic heat drop in the turbine in B.Th.U./lb. (kcal./kg.). It is more usual, however, to make use of a quantity which is proportional to the square of this ratio $q = \frac{\Sigma u^2}{h_0} \frac{\text{sq. ft./sec.}^2}{\text{B. Th. U./lb.}}$, called the *Parsons* coefficient, or simply, the *quality figure*. The efficiency curve of multi-stage turbines as a function of this coefficient q approaches its maximum value at a very flat slope, in other words, a large increase in the quality figure only gives a small improvement of efficiency. When the maximum is passed the efficiency usually falls more rapidly than it climbed. The term Σu^2 of the quality figure also gives an idea of the bulk of the machine, consequently, of the price of the turbine. An increase in the value of the quality figure will mean an increase in the number of stages if the diameter is constant, or for a given number of stages it will mean an increase in the diameter. In any case the price of the turbine will be increased. The curve in Fig. 22 is

naturally a picture also of the efficiency as a function of Σu^2, although it does not give the maximum value of the efficiency. Another figure is often found in texts, especially English ones (28), instead of the figure q, $\lambda = z\left(\frac{d}{10}\right)^2 \cdot \left(\frac{n}{100}\right)^2$ which is only proportional to Σu^2. z is the number of stages, d their mean diameter in inches, and n the revolutions per minute. In conjunction with the steam conditions of the turbine which is being investigated, it naturally serves the same purpose as the quality figure q.

The quality figure can only give the value of the efficiency of a turbine if all the other quantities, especially the steam velocity, are properly chosen in the design. If, for instance, in order to increase the quality figure of a turbine for a high steam pressure and small load, the stage diameter is increased, shorter blades would be obtained. Larger flow losses would result and, instead of the expected improvement, the machine would be less efficient. Similarly, a machine with a large heat drop and small output, which already has a large number of stages, might have its number of stages increased still further in order to increase the quality figure; this would result in an excessively long construction, with large inner and outer glands, and therefore large leakage loss, and a decrease in the efficiency might follow. Moreover in turbines of small capacities the disc friction and ventilation losses increasing with the quality figure have greater importance; hence, such machines should not, for high pressures especially, be designed with too good a quality figure.

If these limitations are kept in mind, the quality figure can very well be used for comparisons of turbines. Naturally, no fixed efficiency should be attached to each value of the quality figure. It does appear, however, that for well-designed turbines, of modern and careful manufacture, a certain range of efficiencies corresponds to every value of the quality figure. In principle a clear distinction must be drawn between impulse and reaction turbines. The quality figure, as defined above, means naturally nothing else than a fictitious ratio u/c_o for the entire turbine. This fictitious u/c_o is $\frac{\sqrt{q}}{223.8}$ $\left(\frac{\sqrt{q}}{91.5}\right)$. With the most favourable values of u/c_o given at the beginning of this paragraph for the impulse and reaction turbine the ratio $\frac{q\text{ impulse}}{q\text{ reaction}} \cong 0.37$ to 0.75 is obtained for turbines with equivalent quality figures; tests on actual turbines show this ratio to be about 0.6 to 0.75. The discrepancy is due to additional losses of a turbine which are independent of u/c_o. Thus, a reaction turbine with a quality figure of 18,000 (3000) will give about the same efficiency as an impulse turbine with a quality figure of about 12,000 (2000) provided all other conditions are the same. Fig. 26, showing the efficiency as a function of the quality figure, has been plotted from the results of the steam consumption tests which have been published in recent years for a series of turbines. It can be seen that the shape of the curve is similar to a parabola.

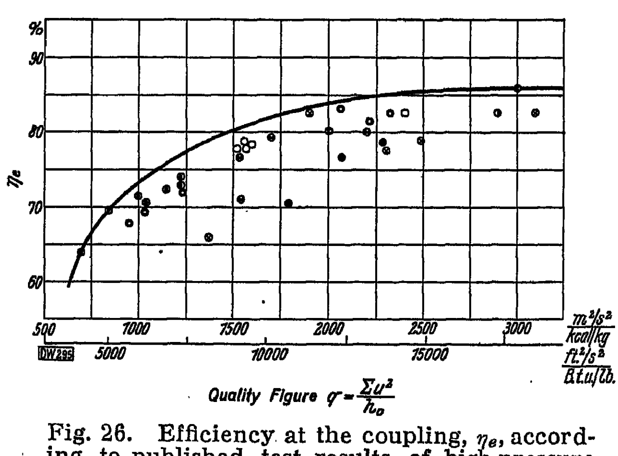


Fig. 26. Efficiency at the coupling, η_e, according to published test results of high-pressure turbines of different makes as a function of the quality figure, q

(28) See *H. M. Martin:* "The design and construction of steam turbines" (London: Longmans, Green & Co. 1913) p. 50.

c. Method of expansion

Frequently, the question is asked which of the two fundamental turbine designs, the impulse or the reaction, gives a better efficiency. This question has no meaning in such a general sense; it can only be answered that each system should be used for the conditions, especially of pressure, for which its qualities best suit it. Further, in modern steam turbine construction there is often no sharp distinction between impulse and reaction types. The majority of designs known as "impulse" work with more or less reaction in the L.P. stages. If the word "impulse" were to be applied only to such turbines as work without any reaction, impulse turbines would be very rare. The term "impulse stage" usually means stages with a small amount of reaction, say, roughly, up to 10 or 15%. This denomination is further justified by the fact that the blade profiles of these stages are the same as for pure impulse stages. In the following the term "impulse stage" will be used in this wider sense.

Theoretically in a reaction blading occur smaller flow losses. The blade efficiency, when calculated from the velocity diagram with fixed velocity coefficients, will be somewhat higher than for an impulse turbine working under the same conditions. The internal efficiency of the turbine is, however, more important than the blade efficiency, and when it is considered, the additional losses, apart from the flow losses, being taken into account, it will be found that the leakage losses are greater in a reaction than in an impulse turbine.

What might properly be called a leakage loss does not exist in an impulse turbine. The leakage across the diaphragm glands can be considered as purely a loss of a certain quantity of steam and, although the weight of the working steam and the output are diminished, the flow across the blading itself is hardly influenced.

The leakage loss in a reaction turbine affects the efficiency in another way. Firstly, it also causes a loss in the weight of the working steam, a certain quantity of steam, depending on the clearance and the pressure drop, flows through the gap without doing work in the moving blades. Secondly, these boundary flows—against the walls of the casing and the inner circumference of the stationary blades—disturb the flow of the neighbouring steam and produce losses by eddies, dispersions and contractions. The blade efficiency itself is affected. The influence is the same, although its value may not be so, whether the moving and fixed blades are shrouded or not. For this reason the radial clearances of reaction turbines must always be kept small, especially in the region of high pressures, where the steam is dense and the blades are relatively short. In the L.P. region, where the volumes are large and the blades long, an ample clearance can be allowed for reaction turbines also. Similar considerations will apply to the case of end tightening as employed in *Parsons'* reaction turbines. If, for instance, an efficiency of 82% could be obtained with a reaction blading with no radial clearance, then, when the clearance is 0.04 in. (1 mm.) and the blade height 0.8 in. (20 mm.), the efficiency will sink by something like 15%, to about 70%. For 0.4 in. (10 mm.) blades it will only be about 56%, according to the most favourable calculations, and will probably be less still. If the loss arising from a given clearance is ignored, the efficiency of a multi-stage reaction turbine depends most likely on the blade length, just as with a multi-stage impulse turbine. For these reasons it is not correct to attribute directly to the use of reaction the superiority which it would appear to have from calculations.

With the impulse system the pressures before and after the rotating blades are the same, or almost so. A loss of steam, arising from a flow from a region of higher to one of lower pressure, can only take place across the inter-stage glands. The flow losses are chiefly reduced to those due to the axial clearance and these are also present in reaction blading. They are caused by the change

in area between the fixed blades and the clearance, and between the clearance and the moving blades. It has been found for impulse turbines that there is an advantage in the axial clearance being small also. The improvement in efficiency to be gained in this way, however, is so small that there is no necessity to reduce the clearance to the safety limit, or even beyond it. The drawing of steam through the moving blades of impulse turbines is harmful also, and can be reduced by keeping the overlap of blades small, also by designing the blading for a small reaction of not more than 15%.

The practice of trying to raise the efficiency by decreasing the axial and radial play, thus reducing the safety factor, cannot be too highly condemned. *The safe operation of a turbine requires certain definite clearances that should not be diminished.* The idea of increasing the efficiency at the expense of the reliability must be considered erroneous.

As a result of this discussion we are led to the following conclusions. For the H.P. region and for small steam quantities, the impulse method is better suited; for the L.P. region and for large steam quantities, the reaction method has the advantage. The largest diameter and the longest blade which may be used without exceeding the maximum allowable stress is smaller for a drum than for a disc. At the present time the first cost of machines plays a more important part than it did a few years ago and favour is again being gained by plain disc-type turbines, with their first stages designed for the impulse method and those nearer the condenser having an increasing amount of reaction. In spite of this, it cannot be denied that pure reaction turbines can be built with a good efficiency also for high pressures, if the steam quantity is large. Furthermore, impulse blading may be used in the L.P. part when small steam volumes, due for instance to a bad vacuum, make it appropriate.

d. Nozzles

The most important parts in a turbine are the fixed and moving blades, which use the kinetic energy in order to convert the potential energy in steam into mechanical power. They must be constructed so that the flow takes place with the smallest amount of loss, they must be able to resist the action of the flowing steam and the high temperatures and withstand the mechanical stress with sufficient safety.

In regard to the shape of nozzle which should be used to get the correct flow, we should be led too far if we entered even into a short discussion on the three principal shapes, with parallel or divergent jet or a continuous discharge. It will only be mentioned that divergent nozzles, which are exclusively for velocities above the critical and are even then only used by certain makers, are probably used in modern designs solely for turbines of small outputs or for velocity wheels. In recent multi-stage impulse turbines, the pitch of the nozzles is chosen as large as the available thickness of the diaphragm will allow. In reaction turbines, the fixed blades are taken as wide as the moving blades, or they may be taken smaller on account of being less stressed, and the pitch is thus determined to a certain extent. Exceptions to this rule are the last stages of large reaction condensing turbines with discs, such as are designed by *B. B. C.* or *S.S.W. - Roeder*. In this case the moving blades are often made narrower than the guide blades on account of the thickness of the disc.

The casting-in of nozzle plates is a method that has often been employed when the steam jet has a sufficient height and when the temperature allows cast iron to be used (Fig. 27). After so many years of experience the methods of casting-in have been so perfected that the steam channels can be depended upon to have sufficiently smooth walls and accurate enough sections and radial heights.

Nozzles of small heights cannot be cast-in with sufficient accuracy and, for this reason, they are often completely machined in the more recent turbines

for instance of the *Lösel* and *Erste Brünner* type. This method of construction not only ensures quite smooth walls, and a reduced surface friction, but it allows the thickness of the diaphragm to be diminished; consequently, the overall length of the turbine can be shortened, and this in turn allows the diameter of the glands and the leakage loss to be reduced. The nozzles are formed by assembling machined parts in which the nozzle channels have been milled.

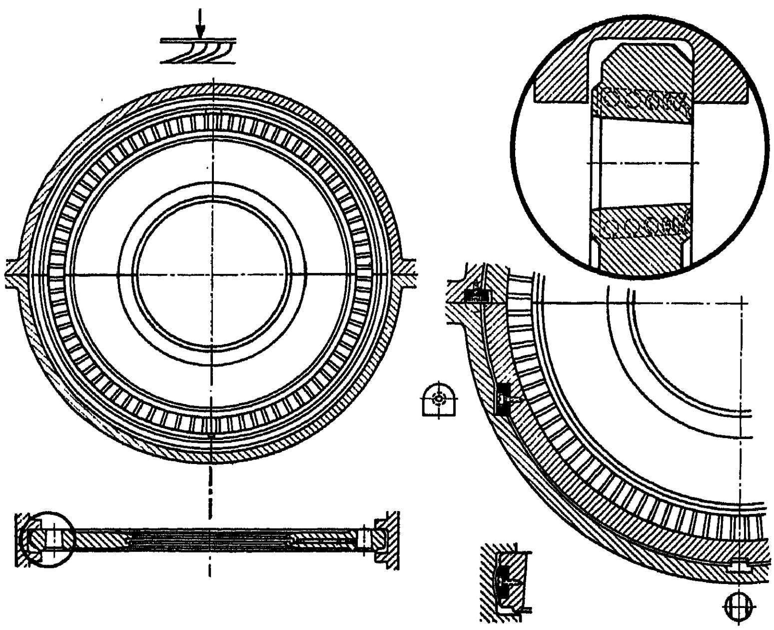


Fig. 27. *A.E.G.*, split diaphragm with cast-in nozzle plates

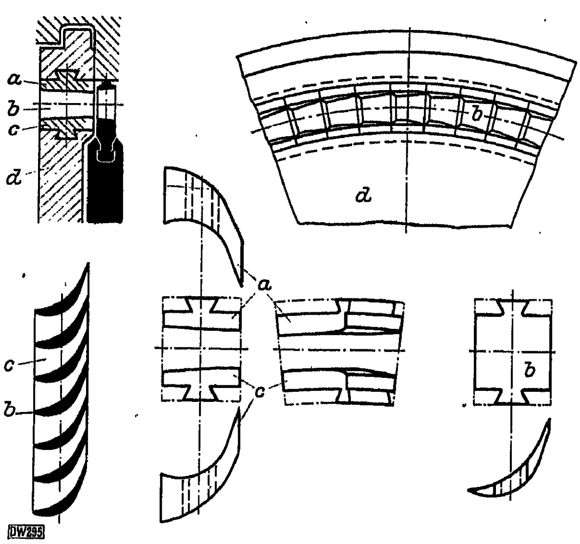


Fig. 28. *A.E.G.*, steam nozzles machined all over

a = Outer distance piece c = Inner distance piece
b = Nozzle plate d = Diaphragm

Another type of nozzle designed for the same purpose is shown in Fig. 28. The nozzles are built up with ordinary steel blades separated by caulking pieces. By this means, channels with smooth walls are also obtained. For nozzles of very small heights, the leakage between the blades and the caulking pieces can be of importance. In that case, it is recommended to braze or weld the nozzles together after assembly. It is possible also with this type of construction, to have a large number of nozzles without getting a badly shaped passage or a worse efficiency. A large number of nozzles gives a good design, as the walls of the nozzles can be chosen thinner and the

3*

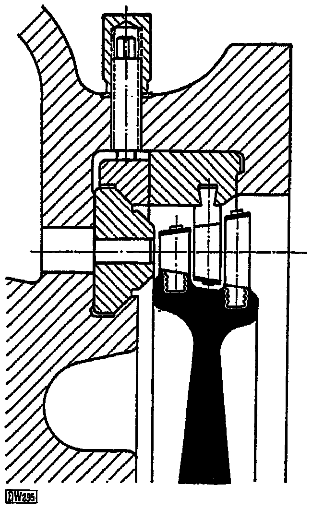


Fig. 29. *B.B.C.*, method of fixing nozzle and guide blade segments to the casing

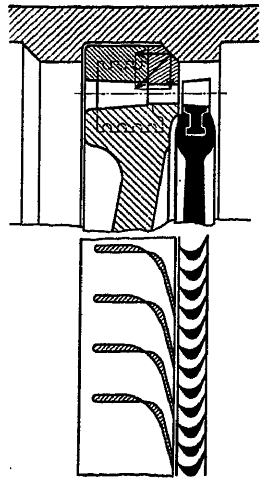


Fig. 30. *B.B.C.*, diaphragm with nozzles machined all over

shock due to the interrupted steam jet is smaller. If the nozzles are designed for continuous discharge, the cause of a periodic steam shock is eliminated. In order, however, not to increase unnecessarily the surface of the nozzle and the friction loss against the walls, the pitch will not be taken smaller, as we have already said, than the thinnest diaphragm required.

Almost every turbine firm has followed its own way in designing and manufacturing nozzles. In Figs. 29 and 30 are shown two different nozzles made by *B.B.C.*. The first type is for superheated live steam and is milled out of segments in special bronze or cast iron, and is closed with a covering ring. These segments are wedged together with a conical shaped ring, which is pressed down by set screws through the casing. The second type is used for diaphragms which have to withstand considerable pressure differences. The diaphragm has plates of steel cast into it, against which are placed nozzle segments with milled blades.

The live steam nozzles of *Westinghouse* (Fig. 31) are made in the same way. The segments are made of two main pieces of cupro-nickel alloy or forged steel. In the first operation they are placed on a special machine where slots are milled for the nozzle plates. These are next placed in position and soldered; the two rings are then welded together. By this means nozzle blocks are obtained with very smooth walls.

Recently, a new method of manufacturing nozzles of small heights for impulse diaphragms has been developed. It was first used in America and was then introduced into Europe by the *A.E.G.*. It consists of brazing the nozzle plates into diaphragms of cast or forged steel in hydrogen furnaces. On

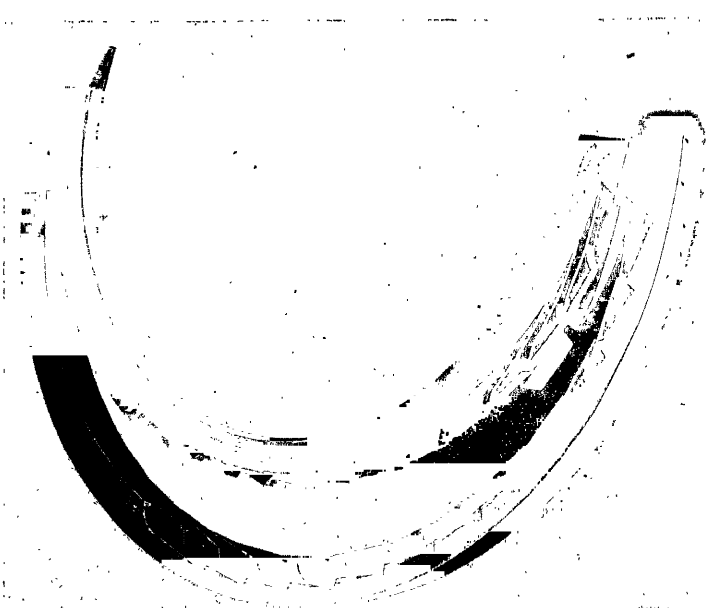


Fig. 31. *Westinghouse*, half rings of a nozzle block without nozzle plates before being welded together

account of their easy manufacture these brazed diaphragms have the great advantage of being cheaper than the built-up kind, without having nozzles any the less accurate, or being any the weaker. As far as can be seen from the relatively short experience of this method, it appears to be successful.

When a multi-stage turbine is designed for full admission, in order to get an overload steam must be let into one or several of the lower stages. This should be done in such a way that when the steam expands from the initial pressure down to the pressure in the stage, the kinetic energy should be used with the best possible efficiency in the moving blades. This can only be achieved to a limited extent, as the steam has a much greater speed when expanding from the initial pressure down to the pressure in the lower stage, than it would have when expanding with the normal stage heat drop. Thus, the value of u/c_0 is too low for the overload stage. There are designs, nevertheless, which have been shown by tests to satisfy the requirements for overload governing.

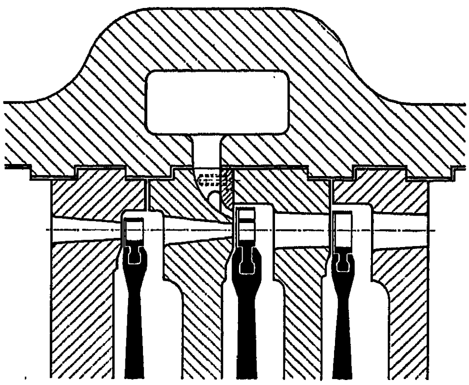


Fig. 32. *English Electric,* overload nozzles

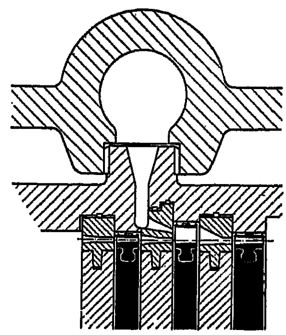


Fig. 33. *Erste Brünner,* overload nozzles

This applies, for instance, to the design of the *English Electric* shown in Fig. 32. The ordinary and the overload nozzles are formed by casting nickel steel plates into the cast iron diaphragm, and the overload nozzles are covered by bolting on a separate ring. The moving blades are made with a shoulder which together form a complete ring separating the overload from the main steam flow.

The overload nozzles of *Erste Brünner* (Fig. 33) are machined out of the solid in a similar way to their ordinary nozzles. There is no partition in the moving blades between the two jets flowing at different velocities. Overload nozzles with very accurate and smooth walls can also be made by the brazing method which has just been described.

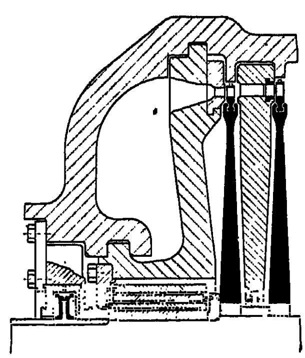
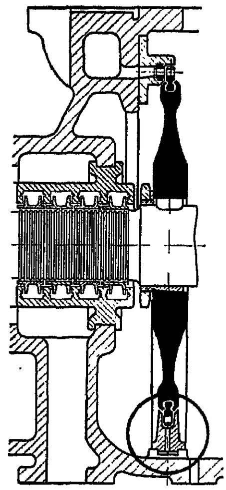


Fig. 34. *Amer. G.E.C.,* steam belt for high pressures

The efficiency of nozzles is not only affected by their form, type and accuracy, but also by their arrangement. The ventilation loss, which can be calculated by the formulae of *Stodola* (29), *Forner* (30), or by the recent formula of *Hodgkinson* (based on the displacement work of the blades) (31), decreases with the ratio of the arc of admission to the total circumference. Hence, it should be endeavoured to

(29) 6th (German) Edition p. 165; English Edition p. 200.
(30) See *Stodola* 6th (German) Edition p. 166; English Edition p. 201.
(31) The Electric Journal 21 (1924) p. 508.

employ full admission. This is facilitated by the present tendency of power stations to build their turbines as much as possible for base loads with simplified regulation. The easiest way to obtain full admission is to employ throttle governing, as is done in the design of the *Amer. G.E.C.* shown in Fig. 34. The nozzle blocks have been carefully cast and placed side by side so that nearly full admission is possible. They are bolted to a cover which is inserted in the casing so as to form a complete annular steam belt. The cover is free to expand without deforming the casing.



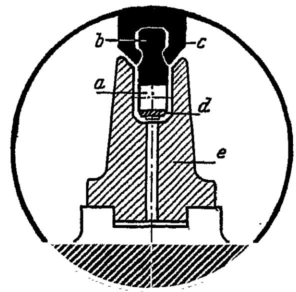


Fig. 35. *A.E.G.*, guard for stages with partial admission

a = Moving blade
b = Distance piece
c = Wheel
d = Coverband
e = Guard

When the governing is done by nozzle control, and partial admission has to be used, the nozzles should be placed, always for the same reason, along as great an arc as possible without obtaining too short blades. At the same time, the separate nozzle groups should be placed near to each other all round the circumference. This leads to several different ways of passing the steam into the casing. For a small admission (i. e. a small arc with nozzles) the nozzle box may be introduced from outside through a hole in the casing. For a large admission, or a large arc with nozzles, the nozzle boxes will be put into place from the inside so as not to weaken the casing with unnecessarily large holes, and they will be bolted outside to a flange on the valve chest. In the case of full or nearly full admission, the steam belt may be cast

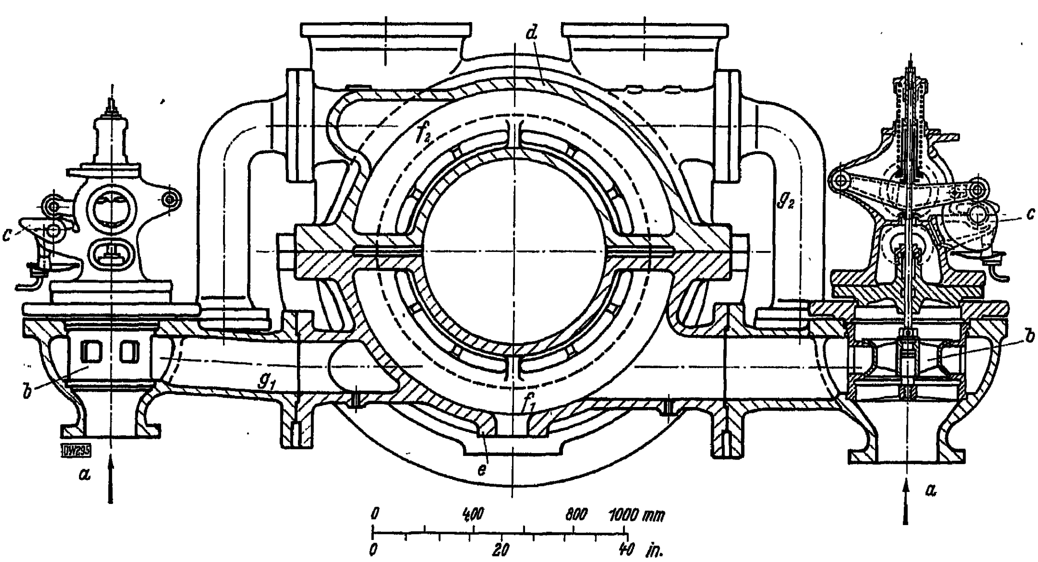


Fig. 36. *A.E.G.*, nozzle groups and throttle valves of a two-cylinder high-pressure turbine

a = From stop valve and steam strainer
b = Regulating valve
c = Governing cam
d = Top half of H.P. casing
e = Bottom half of H.P. casing
$f_{1,2}$ = Nozzle groups 1 and 2
$g_{1,2}$ = Steam passages to the nozzle group 3 and the overload nozzle group 4

in the casing. This should only be done, however, if the initial steam temperature is not liable to cause any dangerous heat stresses in the casing.

The detrimental effect of partial admission is diminished and the ventilation and displacement losses of the blading are decreased when the idle running portion of the blades are protected by a special guard (Fig. 35). Needless to say, such a protection is always used for velocity stages with two or more rows of moving blades.

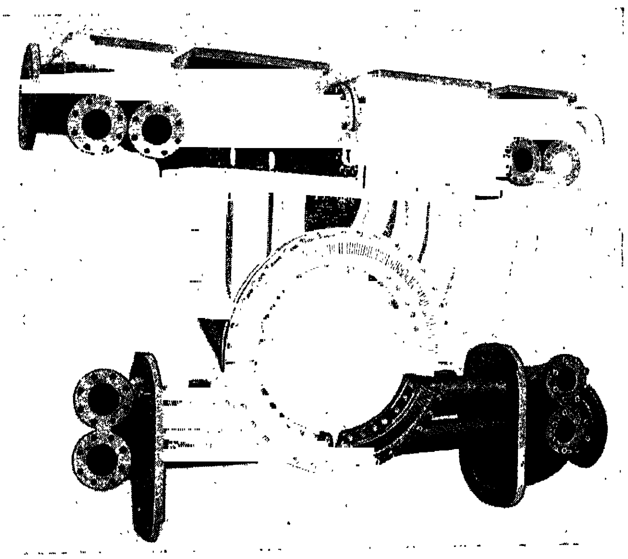


Fig. 37. *A.E.G.*, valve chests and nozzle groups of a combined live steam and accumulator steam turbine for 12,000 kw.

The arrangement of the nozzle groups and valves used by the *A.E.G.* for large condensing turbines is shown in Fig. 36. Full load is obtained with three of the nozzle groups, the fourth group is for the first overload. In most cases there is also a by-pass to a lower stage to give an additional overload.

When frequent changes in load are expected there must, naturally, be a greater division of the groups. This is especially the case when high-pressure turbines are to be used for taking the peak load, or in the case of industrial turbines with several governing gears. In such cases the steam chests and regulating valves are often placed on the H.P. turbine casing. They should, of course, be mounted so that the heat expansions are not hindered. Fig. 37 represents the valve chests and nozzle groups for a governing arrangement of this kind. It is for a turbine working with both live steam and steam from an accumulator. As can be seen, the steam from the two sources is admitted to the first stage through separate valves and nozzle groups. The latter nearly cover the complete circumference.

e. Moving blades

In the designing of nozzles and fixed blades it is usually easy to satisfy the conditions of sufficient strength, and these do not, as a rule, influence the form of the steam path determined by the shape of the channel. In the case of moving blades, on the other hand, rotating at high speed, especially in the L.P. part, the forces are so great that, in order not to exceed the practical limit of stress, they have to be considered just as much as the conditions of flow, and they largely determine the shape of the steam passage. As for every correctly designed element of construction, a compromise must be made between the various opposed requirements, which are here those of strength and a good steam path.

In modern turbines the use is continued of well-known profiles which have been often tried in practice. This has happened in impulse blading with the so-called "limit blade", where the steam channel has approximately an equal depth, and with the "plate blade" made of curved plate of constant thickness, where the steam passage first widens and then contracts on leaving the blade. In reaction blades the usual shape of reaction nozzle has not been

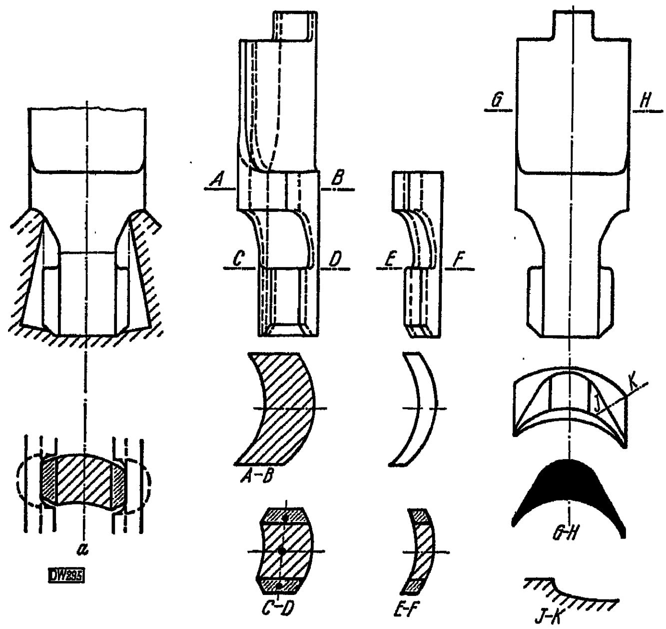


Fig. 38. *A.E.G.*, moving blade with root thickened on both sides
a = Opening for introducing the blade into the groove

essentially altered, with its narrowing steam passage towards the exit, where, however, there is usually no parallel guiding. These original forms have to be adapted to the ever-increasing limit capacity. An example of a modern limit impulse blade for H.P. and I.P. stages of the *A.E.G.* is given in Fig. 38. The blade root is thickened on both sides. From this root it has been milled into a tapering shape; still between two adjoining blades a narrow caulking piece is required.

The blading of the last stages of turbines for limit capacities is one of the most important questions of present day turbine construction. As already mentioned, the first difficulty in design is the question of strength, which requires the blades to be tapered towards their tips. Secondly, there is the divergence of the long blades, the distance between them increasing in proportion to the diameter. The result is not only a distorted velocity diagram (refer to Fig. 10) but also a less effective guiding of the steam between the too widely separated blades. The steam is flung outwards towards the larger empty passage, and this flow can be prevented only, to a certain extent, by a carefully thought out design. A means of decreasing the stresses from the centrifugal force is not only to taper the blade section, but to diminish also its width, or axial dimension, between the foot and the tip. In order to have an equal axial clearance for the whole blade length, the guide blade is shaped to fit; it is wider at the outer than it is at the inner end. A blade of this type is very strong, but great care is needed in the design of the steam path, and an expensive method of manufacture should be expected. It should be remembered that the last stages, on account of the large diameters and high velocities, use a large proportion of the total heat drop, and it should be endeavoured to obtain as good an efficiency as possible for them. Limit turbines will otherwise have a worse steam consumption than smaller machines.

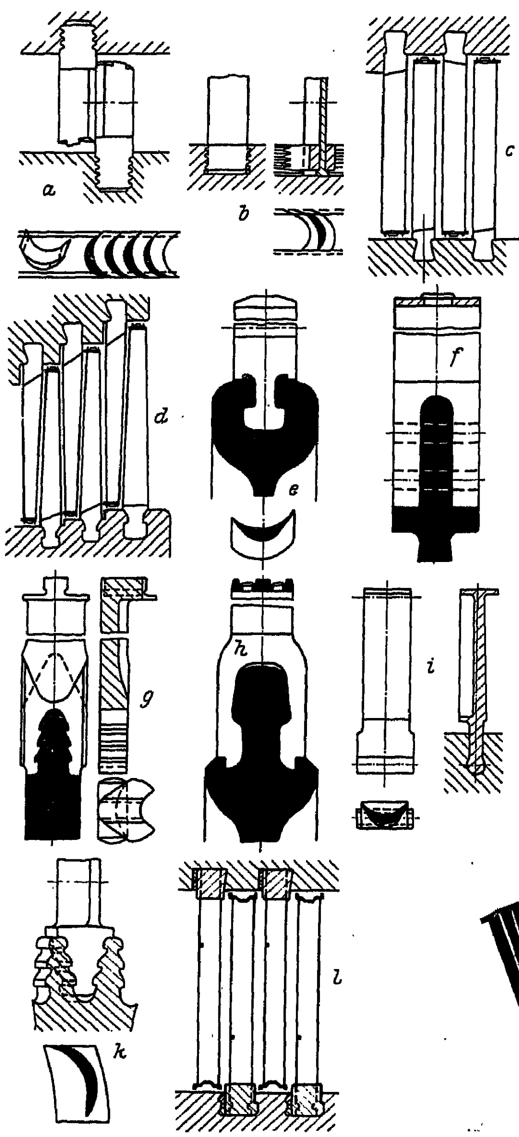
On account of the large number of blades of peculiar shapes which are needed, condensing turbines for limit capacities are a difficult manufacturing problem. Every turbine works has, naturally, evolved its own methods, but these, on the whole, are very similar. As an example, views of a blade of the *A.E.G.* which show the principal stages of manufacture are given in Fig. 39. The reaction blade in question is both tapered and twisted, and a drawing of it is given in the middle of the figure. The rough bar is first rolled or drawn into a profiled rod (Fig. *a*) which must be considered the raw material for the turbine shops. The manufacture begins with the milling of the blade root, which is done simultaneously on both sides with profiled cutters (Fig. *b*). After each operation the accuracy is controlled by means of gauges, which may be seen on the illustrations. In the next stage the taper of the blade bar is milled with an

used by *Parsons;* it is a type which has been applied elsewhere in many forms, as, for instance, in the *B.B.C.* fixation in Fig. *b*. The particular feature of this last type is that the blade section is constant, or almost so, for its whole length. In order to transmit the centrifugal force to the distance pieces, the body of the blade is widened at the foot. The distance pieces are held in the disc or drum by grooves. The advantage of this fixation is that there is no weakening of the blade section, and there are also no obstructions to the flow, as occur in milled blades, when passing from one section to another.

The ordinary dovetail fastening is shown in Fig. *c*.

In Fig. *d*, a limit size of blade is shown. It is fixed by means of a T-shaped root. The root is widened at both sides to reduce the bearing pressure at the flanks and the bending stresses at the neck. In order to avoid an eccentric pull, which would produce high bending stresses, great care must be taken that the line of the centres of gravity of the blade and root sections should be perpendicular to the main shaft and pass through its centre line. The centre of gravity of the two bearing faces is also determined so that any undesirable turning moment is avoided. For very long blades and high centrifugal forces, sometimes a double T-shaped root is used. The T's are either arranged side by side or one above other, as in Fig. 41.

The arrangement in Fig. *e* has a peculiar feature. The T-shaped root is fixed in a slot of the same shape. The blade has, also, a projection on each side which fits over



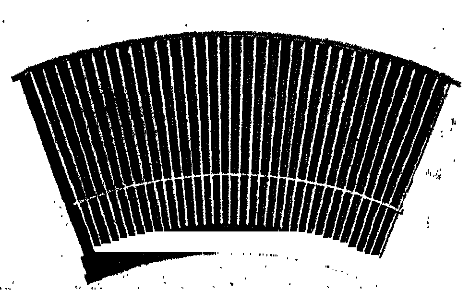


Fig. 42. Methods of attaching moving blades

a = Serrated groove *(Parsons)*
b = Serrated groove *(B.B.C.)*
c = Dovetail groove
d = T-slot *(A.E.G.)*
e = T-slot *(Ford)*
f = Riveted straddled foot *(Brit. G.E.C.)*
g = Straddled pine tree foot *(Ljungström)*
h = Straddled T-shaped foot *(English Electric)*
i = Axial cylindrical groove *(de Laval)*
k = Axial pine tree groove *(Westinghouse)*
l = Base ring segments with cast-in plate blades *(Allis-Chalmers)*
m = Segment of blades of type *l* *(Allis-Chalmers)*

a shoulder on the rim of the disc and prevents the two lips from opening out.

In the design of the *Brit. G.E.C.* (Fig. *f*) the moving blades are fixed to the wheel by a fork-shaped root with counter-sunk rivets. The blades are milled out of a solid rectangular bar and no distance pieces are needed. For short blades a separate coverband is not required, as the shrouding and the distance pieces are in one with the blade, like the design *de Laval* has always used.

Whilst the types *a, b, c* and *f* are only for small or moderate centrifugal forces, the T-shaped root can be used for the highest stresses. The so-called pine tree type of root is probably a development of the T-form. Fig. *g* shows this type, but for the case where the pine tree has been cut out of the rim of the wheel and the blade fits over it. This same reversion of position can, naturally, be made also in the case of the ordinary T-shape, as shown in Fig. *h*. Special precautions have to be taken to prevent the blades becoming loose. When highly stressed the two sides of the fork tend to separate and they press against two ridges on the rim of the disc. This design is similar to that of *Bergmann*.

All these types differ almost only by the shape of the groove and have been inspired by *Parsons*. In a similar manner, the original *de Laval* side-entry fixation (Fig. *i*) has since found a successor. *Westinghouse* has developed a serrated root for the longest blades of large turbines. Like the *de Laval* fixation, its groove is not at right angles to the centre line of the turbine, but is almost parallel to it (Fig. *k*). A fixation of this kind has the advantage that every blade can be put into place, or removed, separately. This benefit is gained, however, at the expense of a higher cost.

Finally, the method of attachment of reaction blading as employed by *Allis-Chalmers* may be mentioned. The blades are joined into sections, as shown on Fig. *l*. The blades are held in a jig and, if a coverband or lacing is used, they are brazed into place. A foundation ring is then cast round the roots of the blades and the sections are fitted into the grooves of the rotor (Fig. *m*).

If blades are held in position by having their roots inserted into grooves, they are generally placed or twisted into these grooves at one point of the circumference. A special piece will, therefore, always be required for closing up the gap by which the blades were introduced. This piece may be composed of several separate parts, as in the case of *Westinghouse* (Fig. 43), or it may be in the form of a copper locking piece which is driven into the gap over a steel wedge, as is used by the *A.E.G.* (Fig. 44). Many other solutions are, naturally, known and have been used with success.

The tip of the blade usually has to seal the clearance in some way. In impulse turbines, the leakage loss being relatively small, the sealing is usually done in the axial direction, and the coverband of the blades projects towards the nozzles. The tightening of the very small clearances in reaction turbines is done in a radial or an axial direction. *B.B.C., Erste Brünner* and many other firms use, at least in the case of blades that are not too long, radial tightening by means of the sharpened tips of the blades. *Parsons,* on the other hand, prefers, even at present, to use axial sealing, especially for high temperatures. Fig. 42 *a* shows this system of "end tightening" by means of a projecting coverband and widened caulking pieces. This design will certainly give a good steam flow, there will be no friction between the jet of steam and the steam casing, as there is in case of radial sealing. However, the axial position of the rotor has to be very carefully adjusted and this has even to be done, sometimes, when the machine is running. The *English Electric* have adopted this method of *Parsons*. They have recently given up using the steel coverband itself for tightening, but they

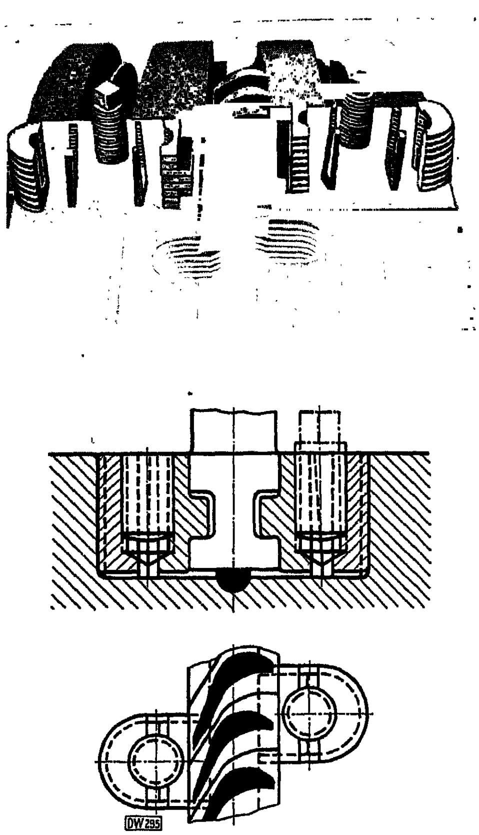


Fig. 43. *Westinghouse,* closing piece of a row of moving blades

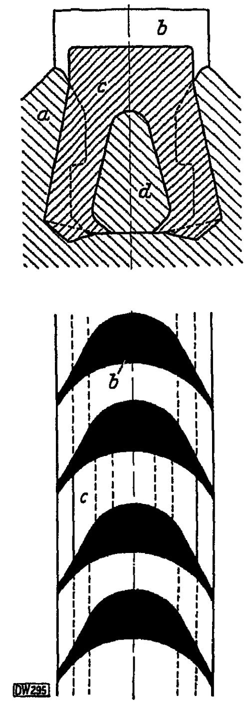


Fig. 44. *A.E.G.,* locking piece of a row of moving blades

a = Blade carrier
b = Moving blade
c = Copper locking piece
d = Steel wedge

rivet into it a copper strip (Fig. 45) which projects over the entrance edge of the blades and seals against the stationary parts. If the two surfaces touch, the copper wears away and the blades themselves are undamaged.

Another way of keeping the leakage loss down is used by *B.B.C.*, the so-called bridging-over of the clearance (Fig. 46). The sequence of coverbands and distance pieces of the fixed and moving blades overlap each other like tiles. This system has also been used for a long time

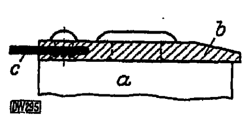


Fig. 45. *English Electric,* sealing of axial clearance by means of bi-metal shrouding

a = Moving blade
b = Coverband of S.M. steel
c = Copper strip, 1/32 in. thick

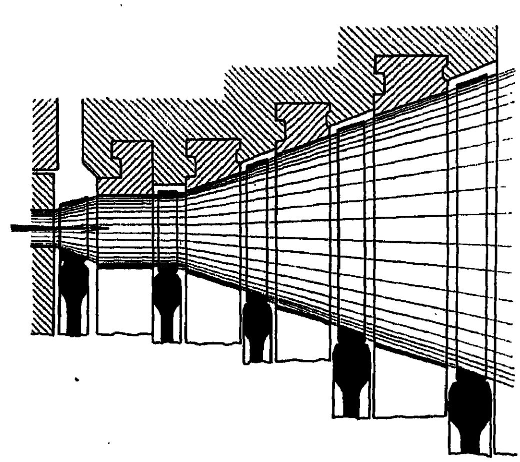


Fig. 46. *B.B.C.*, bridging-over of the clearance by means of coverbands arranged in steps

by other manufacturers, especially for the velocity type of impulse stage. There are neither diaphragms nor interstage glands, the guide blades have coverbands much like the moving blades. The entrance edge of the nozzle is nearer the centre at its inner end than is the corresponding point of the leaving edge of the previous moving blade. There is what may be called a positive overlap and the steam in the space between the two adjoining discs is sucked into the nozzle by the jet. The outlet edge of the nozzle overlaps at its inner end the inlet edge of the following moving blade; the overlap is here negative, the inner stream of the jet leaving the nozzles strikes against the distance pieces of the rotating blades, there results a local compression and the steam between the discs is pushed back. For these reasons it seems likely that a mean pressure, between those in the clearances, before and after the guide blades, is established in the space between the two discs and it would limit the undesirable leakage of steam. Tests for different blade lengths would enable a comparison to be made with the overlaps usually allowed in practice. Up to the present, however, no results have been published.

Fig. 47. Monel Metal lacing wire soldered to stainless steel blade with silver solder. Microphotograph, about 1.5 times full size

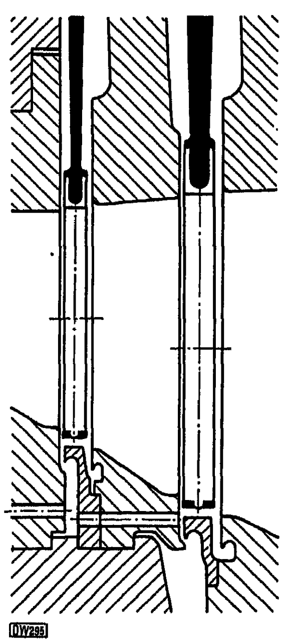


Fig. 48. *Brit. G.E.C.*, water collecting and drainage channels in the last stages of condensing turbines

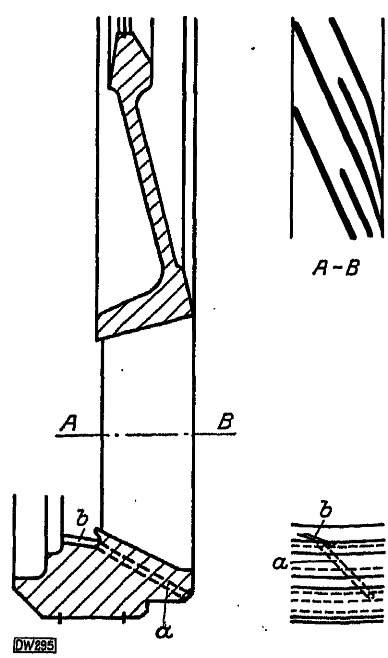


Fig. 49. *Escher Wyss*, diaphragm with drainage holes

a = Drainage hole
b = Baffle plate

The purpose of the coverband is not only to guide the flow, it is also useful for stiffening the blades, especially long ones, against each other and, at the same time, it prevents vibrations. If no coverband is used, or if a further stiffening is required for long blades, one or more lacing wires may be employed. These wires are soldered or brazed to the blades. Soldering was at first difficult in the case of stainless steel for two reasons chiefly. The steel cannot be heated over a temperature of about 1470° F. (800° C.) without losing its strength, also soldering becomes usually all the more difficult, the lower the temperature. Good results can only be obtained if a thorough inspection is made and the workmanship is both skilled and careful. Fig. 47 shows that the operation can be done perfectly. A stainless steel blade has been joined to a Monel Metal lacing wire. The silver solder has taken everywhere and the steel has not been overheated.

In connection with the moving blades in the last L.P. stages of condensing turbines, another precaution has to be mentioned. It is for extracting the water which condenses in the expansion below the saturation point, and protecting the blades from wear. The solution of the *Brit. G.E.C.* (Fig. 48) simply collects the water which is hurled towards the periphery, and leads it, through holes in the diaphragms, to the condenser. *Escher Wyss* also use holes round the circumference to drain away the water (Fig. 49). In order to improve the performance, baffle plates are inserted near each hole to guide the water away. Other designers use slightly different arrangements, based on the same principle. The practical value of the method has not yet been determined. The lengthening of the life of the L.P. blading to a considerable degree would completely justify any efforts.

f. Casings

The H.P. casing is always relatively small, and it only has an awkward shape if the governing is complicated and many flanges are needed. This may happen especially in the case of industrial turbines. Usually the operating temperatures

Fig. 50. *B.T.H.*, bearing pedestal with inclined pads to carry the H.P. steam casing

and pressures determine the design. One of the most important requirements is that the casing should be able to expand freely when it is heated up, and the alignment of the set should not be affected. This condition must be fulfilled by the methods of connecting the casing to the bearing pedestal. The best

known method of connection is by means of semi-circular facings on the bottom half of the casing and the bearing pedestal. Two horizontal keys, close to the joint in the casing, and a vertical key, at the bottom of the casing, allow the heat expansion to take place in a radial direction from the centre line. The *B.T.H.* use pads, inclined at 45°, on the bearing pedestal and the bottom part of the casing (Fig. 50). When the casing expands at right angles to the turbine centre line, it slides on these pads, and the position of the centre line, relative to that of the rotor, does not change.

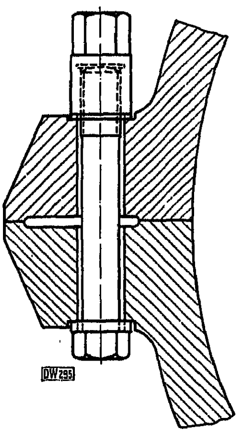


Fig. 51. *Brit. G.E.C.*, tightening of the horizontal joint of the steam casing

Another difficulty is often the tightening of the horizontal joint against the pressure. Some makers use pure metallic joint, others use asbestos cord or similar materials. The method shown in Fig. 51 may be noticed.

The greatest requirements for strength occur, naturally, in casings for turbines using very high pressures. Further details of construction will be given in this connection later.

All designs are greatly influenced by the materials; hence, the shape of the H.P. casing is quite different if it is made of cast iron, of cast steel or of forged steel. As cast iron is no longer used for higher temperatures, the only suitable materials which are available at present are cast steel and forged steel, and it looks as if cast steel will become universally employed.

The materials for exhaust casings, on the other hand, have never been a difficult problem. Cast iron has always been used. There are other questions, however, which complicate the design of exhaust casings. As already stated, the following conditions have to be fulfilled:—

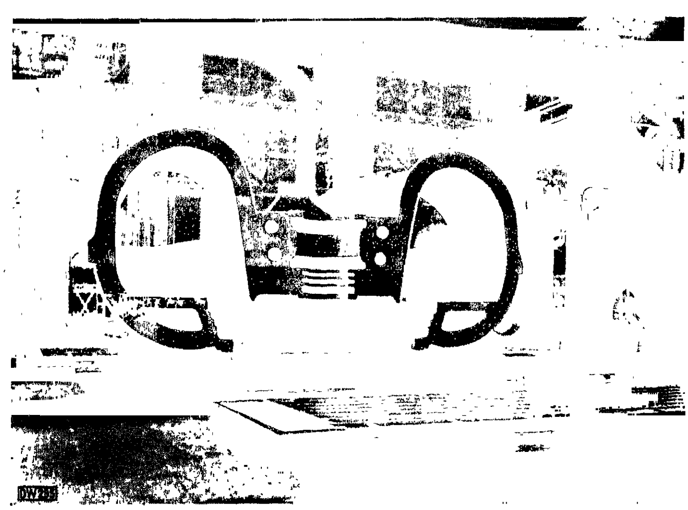


Fig. 52. *A.E.G.*, lower part of the exhaust casing of an 80,000 kw. turbine

The transformation into pressure of the kinetic energy of the steam leaving the blading must begin as near as possible after the last wheel; it must take place as quickly as possible, yet it must be gradual.

The curvature must be gentle, the area of the steam passage in a direction perpendicular to the flow must increase continuously, causing the steam velocity to decrease gradually.

The jets of steam escaping from the last wheel must not hinder each other.

As few guiding surfaces as possible should be provided in the exhaust casing, so as to keep down the friction losses against the walls.

These conditions are totally or partially fulfilled in nearly all turbine designs. The most noticeable characteristic of an exhaust casing built on these lines is the so-called diffusing guides which separate the steam coming out of the top half of the last wheel from that out of the bottom half (Fig. 52).

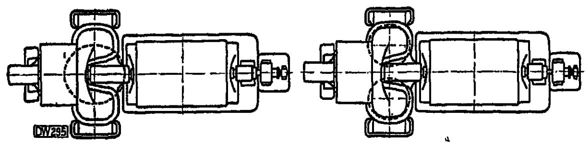


Fig. 53. Turbo-alternator. Single-casing steam turbine with one (left) or two (right) exhaust openings

Fig. 53 gives diagrammatically the two designs which are the most frequently used for single-flow turbines. In the first case there is only one connection to the condenser. In the second case there are two exhaust openings, which may be joined to one condenser situated across the set, or to two condensers parallel to the set.

The most common types of exhaust casings for single-cylinder turbines are given in Fig. 54. Sketch *a* shows the diffuser-shaped type of exhaust branch of the *A.E.G.*. The section of the steam passage increases very gradually from the last stage down to the condenser. In the design *b* more importance is attached to placing the condenser as close as possible to the last stage than to obtaining a well-shaped diffuser. Type *c* can be used for large volumes of steam,

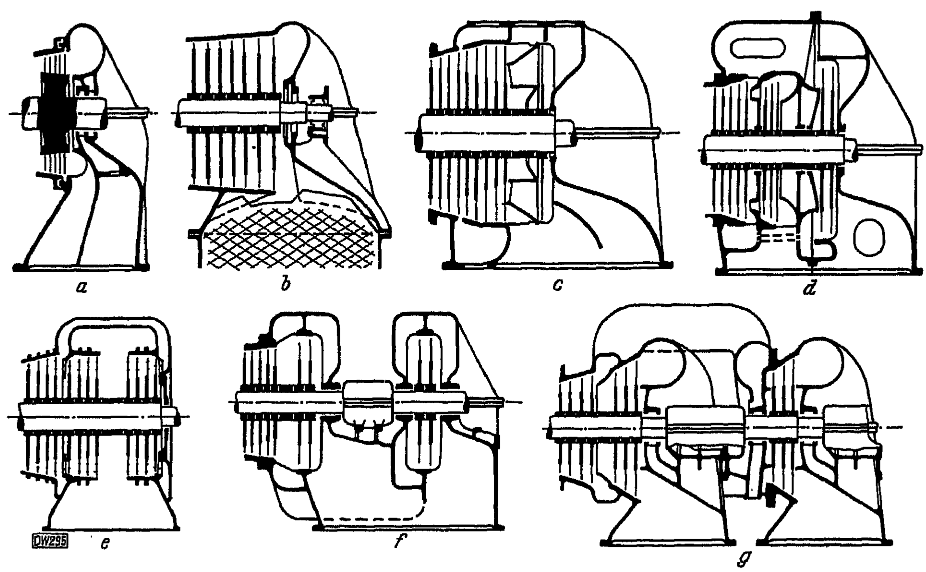


Fig. 54. Types of exhaust casings with unsymmetrical L.P. stages, mostly for single-cylinder turbines

a = *A.E.G.*, diffuser shape
b = *Metro-Vick*, condenser close to the blading
c = *Metro-Vick*, multiple exhaust
d = *Brit. G.E.C.*, double parallel-flow
e = *Ford*, double counter-flow
f = *Brit. G.E.C.*, double parallel-flow with intermediate bearing
g = *Bergmann*, double parallel-flow with a second separated L.P. casing

as in turbines for limit capacities. This shape is used by both *Metro-Vick* and *Westinghouse*. The steam flow is divided in the last stage by a partition across the fixed blades. The steam in the outer ring expands down to the exhaust pressure. The pressure drop in the inner ring is relatively small, as the leaving angles of the stationary blades are larger here. The steam in this inner portion passes through the moving blades, with another small pressure

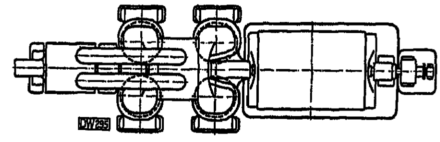


Fig. 55. Turbo-alternator. Two-casing steam turbine with double-flow L.P. end and four exhaust openings

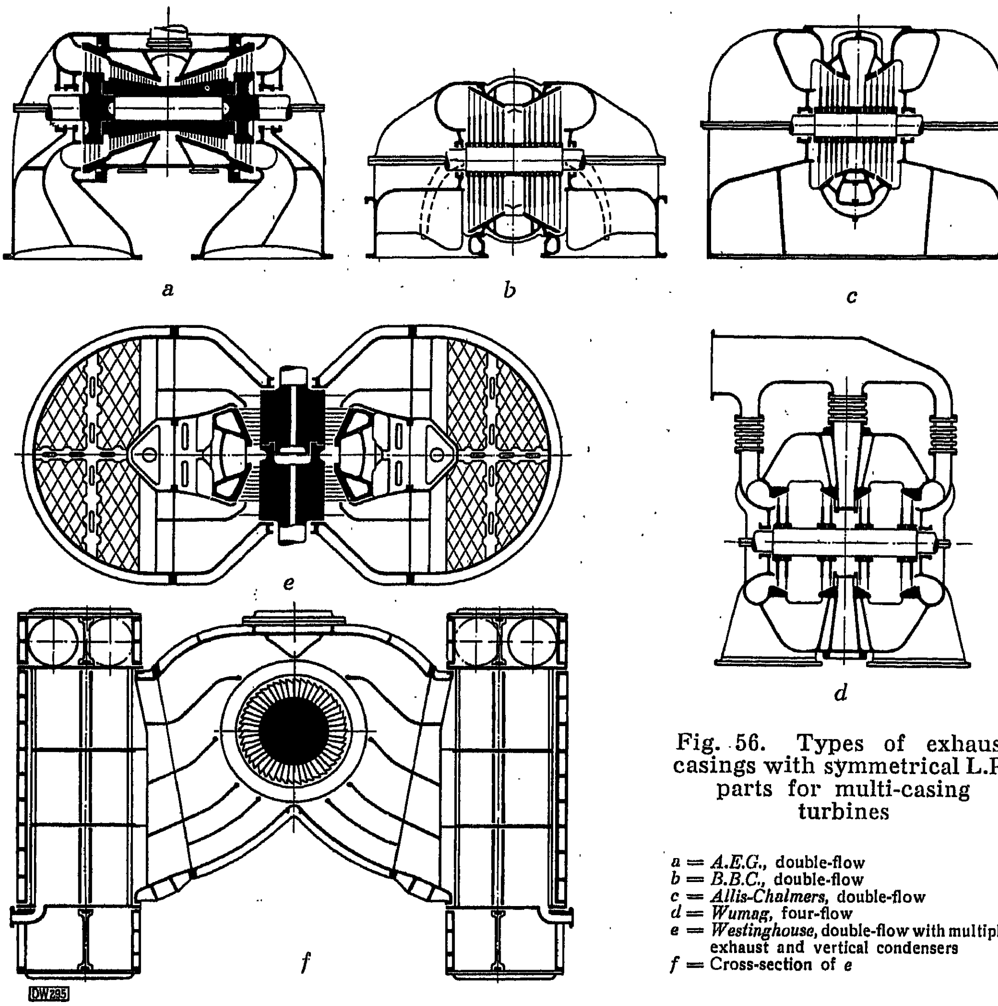


Fig. 56. Types of exhaust casings with symmetrical L.P. parts for multi-casing turbines

a = *A.E.G.*, double-flow
b = *B.B.C.*, double-flow
c = *Allis-Chalmers*, double-flow
d = *Wumag*, four-flow
e = *Westinghouse*, double-flow with multiple exhaust and vertical condensers
f = Cross-section of e

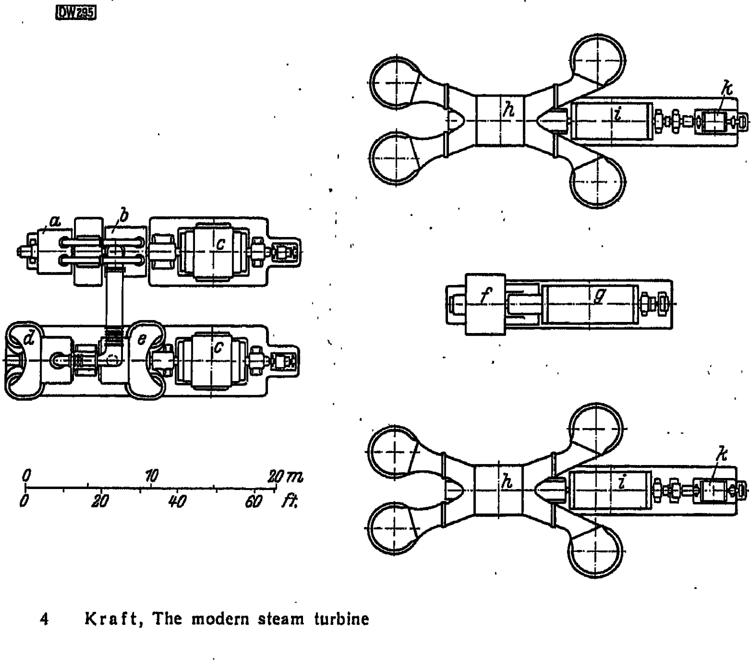


Fig. 57. Four-casing *A.E.G.* high-pressure turbine for 80,000 kw. at 1500 R.P.M. (left) and three-casing *Amer. G.E.C.* high-pressure turbine for 208,000 kw. at 1800 R.P.M. (right)

a = H.P. turbine
b = I.P. turbine
c = Alternator, 40,000 kw.
d = 1st L.P. turbine
e = 2nd L.P. turbine
f = H.P. turbine
g = Alternator, 76,000 kw.
h = L.P. turbine
i = Alternator, 62,000 kw.
k = House set, 4000 kw.

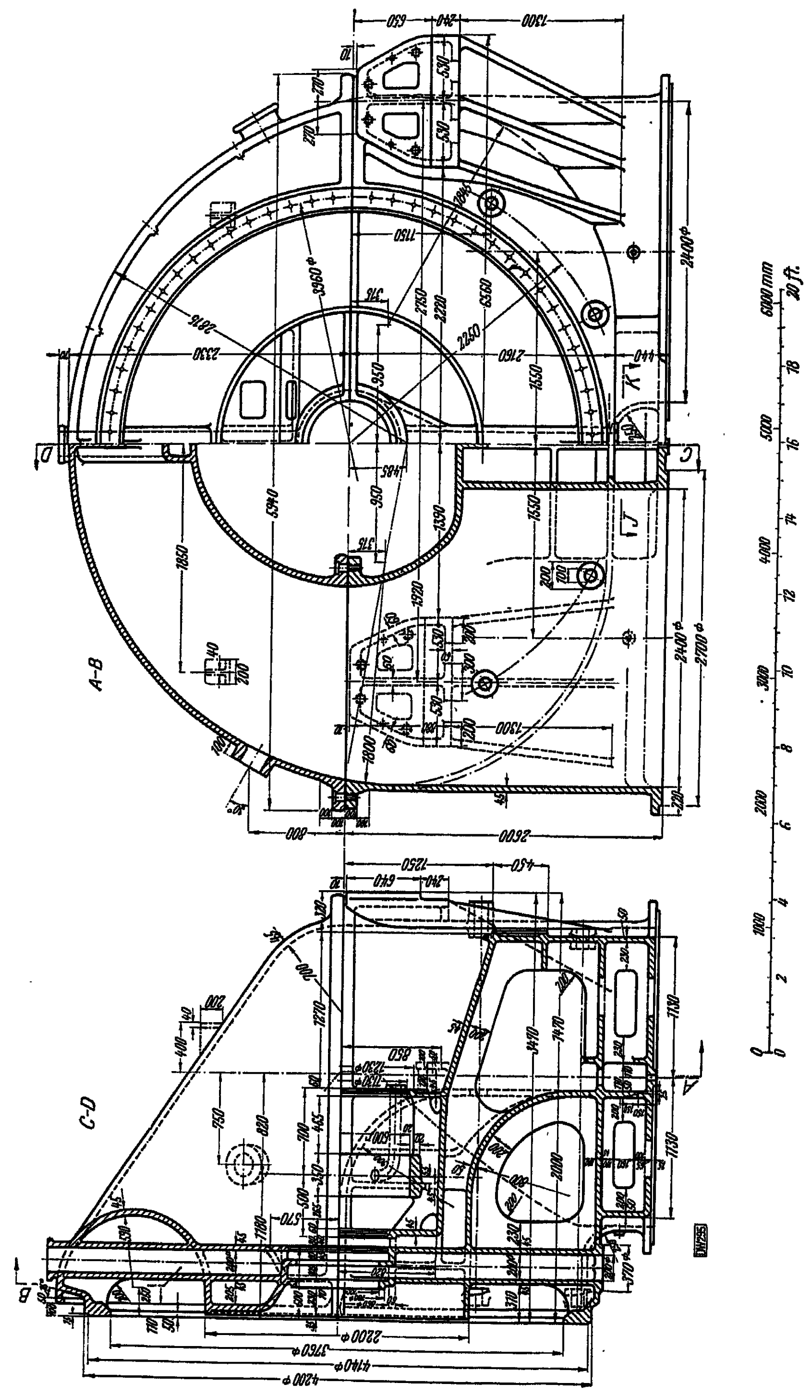
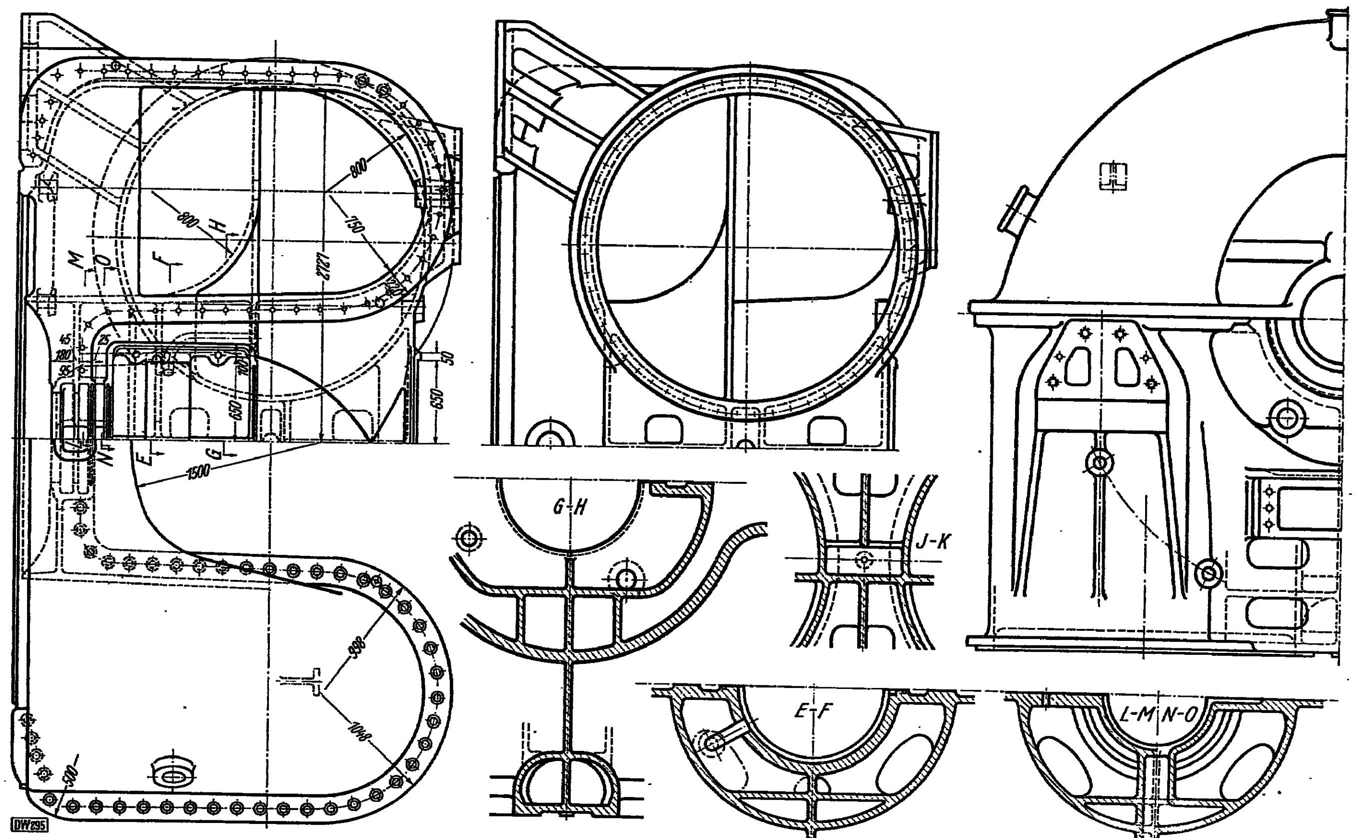


Fig. 58.



Figs. 58 and 59. *A.E.G.*, exhaust casing of a two-cylinder double-flow high-pressure turbine for 85,000 kw. at 1500 R.P.M.

4*

drop, and it is completely expanded in a separate stage. This arrangement of the L.P. stages enables large volumes of steam to be used, at high speeds of rotation and vacuum, without employing unduly long blades, or highly-stressed wheels of large diameters. The efficiency of the turbine is, however, affected. Blades for designs of this kind are shown in Figs. 40 and 41.

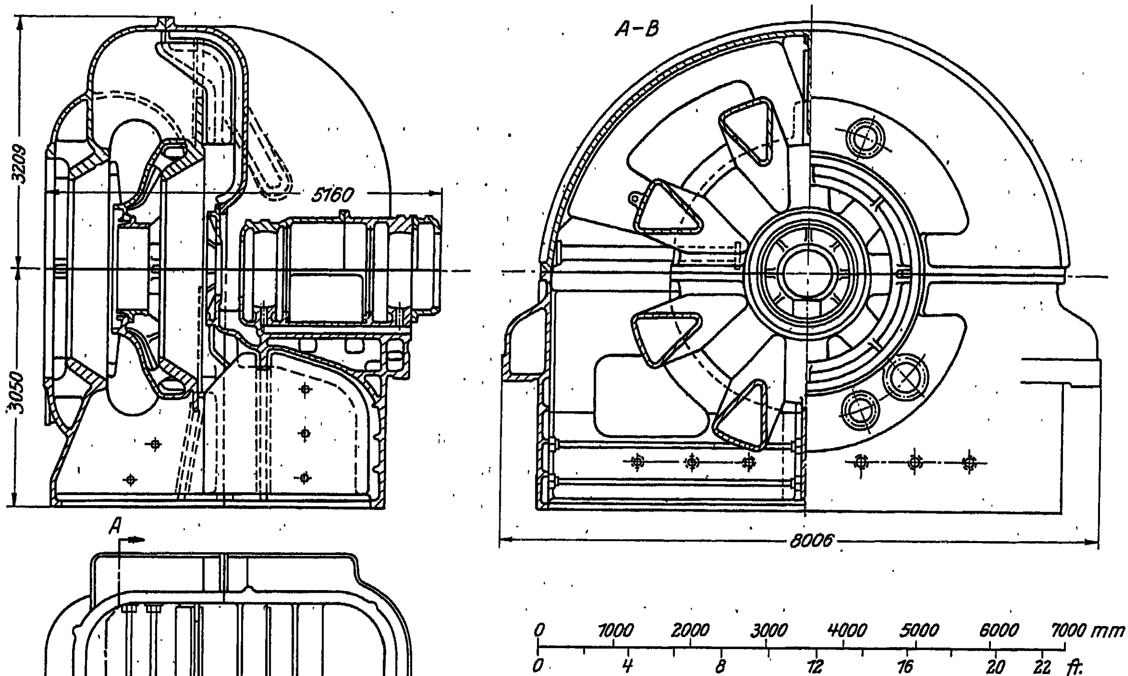
Other turbine makers have employed this same idea in different ways. In the design *d* of the *Brit. G.E.C.*, part of the steam is led directly through three L.P. stages of the same diameter, the remainder expands in two stages of larger diameter, which are placed further away. The two groups of stages have the same heat drop. They are enclosed in the same L.P. casing but are separated by an intermediate gland. The two steam flows in the design of the *Ford Motor Co.*, in sketch *e*, take place in opposite directions, and the steam discharges into a common exhaust branch. The steam going to the farther group of stages flows between the double walls of the exhaust casing. Contrary to nearly all other designs, the makers of these turbines do not attempt to guide the exhaust steam, neither is there a partition to prevent the collision of the two streams. Another feature of this turbine is the radial division of the steam flow in the last stages. A symmetrical arrangement of the L.P. wheels, with flows in opposite directions, has the advantage of balancing the axial thrust of the stages, which usually work with reaction.

A further method of dividing the steam flow in the last stages is shown on another design of the *Brit. G.E.C.* in sketch *f*. The two groups of stages, with each two wheels, are, in this case, separated by two glands and an intermediate bearing, they are in different casings; there is, however, only one exhaust opening. The steam going to the second group of stages is led through wide channels cast in the lower half of the casing. A further development of this same idea is shown on sketch *g*, which is an arrangement of exhaust casings designed by *Bergmann*. The second flow of steam is led through piping to another exhaust casing, separate from the first, and with its own exhaust opening. This is almost a multi-casing turbine, which is a type having at least *one* separate L.P. turbine.

The most usual type of multi-casing turbine, as far as the arrangement of the exhaust openings is concerned, is shown in Fig. 55. The steam is led to the middle of the L.P. casing and is divided into two streams which expand symmetrically in either direction. In this way double the leaving area of a single-flow machine may be obtained. Two or, as shown in the illustration, four condenser connections may be used.

The detail design of the exhaust casing follows the principles given above. The various designs are very similar. Fig. 56 *a* shows the L.P. part of a double-casing, double-flow *A.E.G.* turbine. It may have two or four exhaust openings. The design *b* is similar and is of the *B.B.C.*. The *Allis-Chalmers* design, on sketch *c*, only differs from the two previous ones by having a single exhaust opening. The reason for this is that in America condensers with cast iron shells are preferred, especially for large units, and the shape of the opening may be chosen freely. Sketch *d* is an example of the so-called multiple-exit type. *Wumag* have used twice, in one casing, the idea in the design of Fig. 54 *e*. In this way four times the leaving area of a single-flow turbine is obtained. Thus, for the same speed of rotation and leaving loss, the limit capacity ought, theoretically, to be four times larger.

Large surface condensers are frequently placed vertically in America. This prevents the tubes getting coated too quickly in the case of dirty water, also it facilitates the cleaning operations. An example of such a design is shown on sketches *e* and *f* of a double-flow *Westinghouse* turbine. It can be seen that the exhaust opening is very large, the steam passage is very short



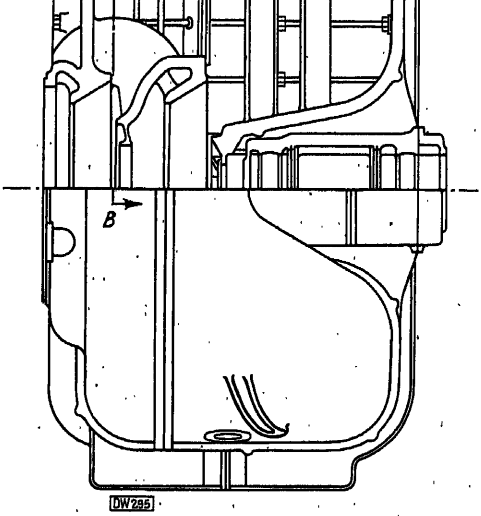


Fig. 60. *Allis-Chalmers,* exhaust casing of a single-casing double-flow high-pressure turbine for 65,000 kw. at 1800 R.P.M. (similar to Fig. 151)

and only slightly curved, and a flow almost without any losses will be obtained. There are, in this case, four separate exhaust openings which lead to two condensers. Each of the two flows in the turbine is divided in the last stage in the manner shown in Fig. 54 *c*, and the streams are kept apart by a guide vane.

These examples show the limits for present day constructions. The unit capacities of turbine plants can, naturally, be increased still further by raising the number of casings, thus practically any output may be obtained. Two well-known examples are sketched in Fig. 57. On the left is the 80,000 kw. turbine of the Klingenberg Super Power Station, near Berlin, with four casings. On the right is the 208,000 kw. three-casing turbine of the State Line Generating Station, in the United States. The first machine has two horizontal condensers and the second has no less than eight vertical ones.

By raising the unit capacities the efficiency is improved, as all the losses are reduced, with the exception of the leaving loss. It is, therefore, better to install in a station a few machines for large outputs, than many small ones. For this same reason, it is an advantage to diminish the number of power stations so as to be able to use large units. The design of turbines for large outputs is, naturally, different from that for normal capacities but, at the present day, turbines of any size could be built if there was a demand for them and they could be sold. In this connection, naturally, other important

factors have also to be considered, such as, especially, the amount and the variations of the load, the necessary reserve of power, and the present limit of the generator capacity.

After the previous diagrammatic sketches, a few details of the construction of exhaust casings are given. Figs. 58 and 59 show the shop drawings of the exhaust casing for a large *A.E.G.* turbine. In Fig. 60 is the exhaust casing of a large single-cylinder double-flow *Allis-Chalmers* turbine (similar to the example in Fig. 151). It is not usually difficult to make designs of sufficient strength, usually the minimum thickness that can be obtained for L.P. casings in the foundry is much greater than is required for reasons of strength. Only in the case where the coupling bearings of the turbine and generator have no separate pedestal and have to be carried by the exhaust casing, must care be taken that the casting is sufficiently strong to withstand the bending due to the weight of the two rotors. Experience and knowledge of casting are needed, however, to design large exhaust casings, easy to mould and cast, and which do not crack in cooling. Naturally, the machine tools and the means of transport which are at hand, or can be procured, should also be taken into account. Thus, the manufacture will not be unnecessarily complicated, and transport by rail to the site will not be made impossible by exceeding the loading gauge.

g. Glands

Besides the losses due to the flow and the clearances in the blading, and the disc friction and ventilation, special attention must be paid to the losses in the outer and inner glands. They are fundamentally important particularly for turbines of small outputs and when the pressure inside the casing is high. In these cases the starting point for the design of the turbine will be the losses in the glands. For condensing turbines a good arrangement, when possible, is for the leakage steam from the H.P. gland at full load to be just sufficient to seal the L.P. gland against the vacuum. This requires the heat drop in the first stage to be chosen all the larger, or the pressure after it to be all the lower, the smaller the output. In the case of back-pressure turbines it should be remembered that the steam working in the turbine is to be used afterwards in a process, and the gland loss can be compared, for instance, to that of a safety valve on the boiler which is continuously blowing-off, or, if the steam from the H.P. gland is led into the process main, the loss is similar to that of a reducing valve which would be always throttling live steam down to the exhaust pressure. Consequently, when designing a back-pressure turbine of small output, all the losses, including the gland losses, should be compared by calculation, and the speed, number of stages, the division of the heat drop and distribution of pressures should be chosen so as to give the minimum of losses. For turbines of small outputs, of condensing or back-pressure types, the number of stages and the quality figure will similarly be kept small so as to obtain a short overall length and moderate shaft diameter.

Every gland consists of two regions, separated by a gap. The main or inner portion, which is the longer, has to seal the internal pressure; the secondary or outer portion, has only to seal the gap against the atmosphere.

The principle underlying the labyrinth-type glands, i. e. the majority of present day glands, is that the steam speed in the clearances must be annulled in every stage of the labyrinth through sudden changes in the leakage area and in the direction of flow. In this way eddies are continuously produced. It has been shown, theoretically and by research, that the gland loss varies, approximately, in the inverse ratio of the square root of the number of the constrictions, hence, by increasing this number, the leakage is reduced (32). The same laws apply

(32) For the theory and calculation of glands, refer to *Stodola*, 6th (German) Edition p. 153; English Edition p. 186..

to the outer glands as to the balance pistons, which are used in reaction turbines to balance the axial thrust. No special reference to them is needed, therefore.

Labyrinth packings are often made by simply cutting grooves in the rotating part, whilst in the stationary part, or the casing, circular fins of an appropriate section are inserted so that they penetrate into the grooves of the rotor (Figs. 61 and 62). The *B.B.C.* type seals either in an axial or a radial direction, the *Parsons* type seals in both directions. The axial clearance varies when the machine is running on account of the different expansions of the rotor and the

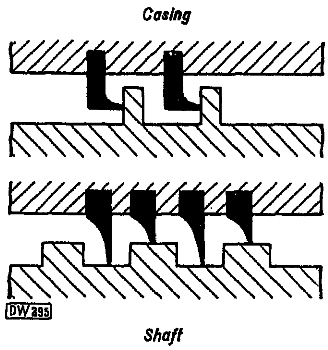


Fig. 61. *B.B.C.*, axial and radial labyrinth packings

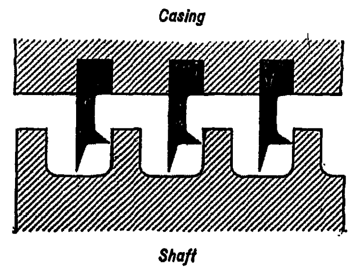


Fig. 62. *Parsons*, labyrinth packing sealing in radial and axial directions

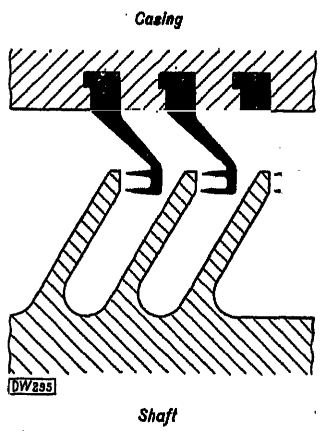


Fig. 63. *Westinghouse*, axial labyrinth packing

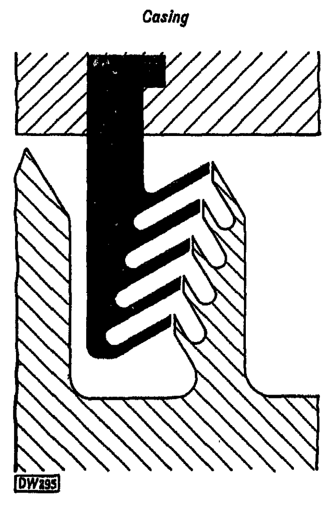


Fig. 64. *Metro-Vick*, axial labyrinth packing. About 1.6 times full size

casing and it may be adjusted by a special device, usually connected with the thrust block. The radial clearance, on the other hand, remains almost constant. The latest form for the stationary fins of *Westinghouse* H.P. glands is probably based on these considerations (Fig. 63). Another type of gland with axial sealing is that of *Metro-Vick* (Fig. 64). The characteristic of both these designs is to allow for the fins a certain amount of flexibility in an axial direction; thus, if the rotor gets displaced, it will not get damaged or distorted by local heating.

According to the opinion of *Parsons*, the flow through radial constrictions should always be from the outside towards the centre so as to be opposed by the

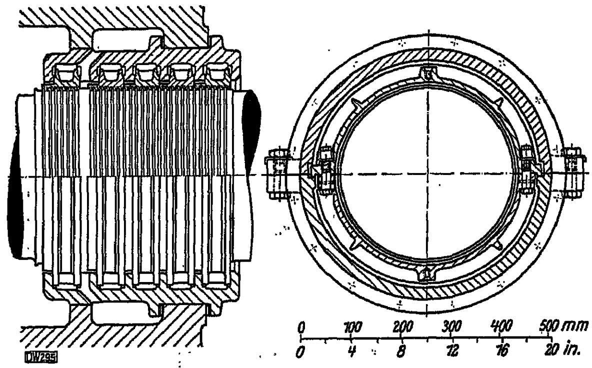


Fig. 65. *A.E.G.*, labyrinth-type packing gland

centrifugal force. Except for very large machines, where more play may be allowed, the usual clearance for labyrinth packings for all designs is between about 0.008 and 0.01 in. (0.2 and 0.3 mm.). Some types even go down to 0.004 in. (0.1 mm.). For reasons of reliability, however, an emphatic warning is given here, once more, against the use of too small clearances.

In the original labyrinth packings of *Erste Brünner* the sealing fins on the shaft are made of thin plate of a heat-resisting alloy, they are fixed by a series of separate steel rings, and these are held in position by a nut. The fins are bent and sharpened at their tip, similar to the edges of the stationary fins, which have an obtuse angle. The idea is that the thin plates ought to bend under the steam pressure and diminish the clearance.

A similar conception for the fixed part is seen in the outer glands of the *A.E.G.* (Fig. 65). In this case also, the fixed fins are grouped together on separate rings out of which they have been turned. The rings are inserted into the boxes and are provided with radial play so that they can follow the deflection of the shaft. On account of the large number of stationary fins the sealing is considerably better than for other types of design.

The H.P. glands of the *English Electric* also have some particular features. It will be especially noticed that the stationary and rotating fins do not intermesh (Fig. 66). There are other interesting points, such as the radial division of the fixed rings into two parts, also the construction of the rotating part out of separate rings.

Labyrinth packings for very high pressures require a great number of stages and become very long. In order to avoid lengthening the machine, the packing may be divided into groups which are superposed so that they follow each other in a radial instead of an axial direction (Fig. 67). This design was first introduced by the *Amer. G.E.C.*. Each labyrinth group, whether it be rotating or fixed, is made in one piece and all the parts must, therefore, be slipped over the shaft. It is for this same reason also, that the fins shown on the illustration do not intermesh. This type of gland gives a very good sealing; to be inspected, however, it not only requires the top half of the turbine casing to be lifted, but also the withdrawal of the rings along the shaft.

Another type of labyrinth which enables a great number of fins to be employed is used in the *Ljungström* radial flow turbine and will be discussed later.

The interstage glands, used in multi-stage impulse turbines to prevent leakages across the diaphragms, are designed on these same principles. To give an example of the many types, which are usually very similar, only

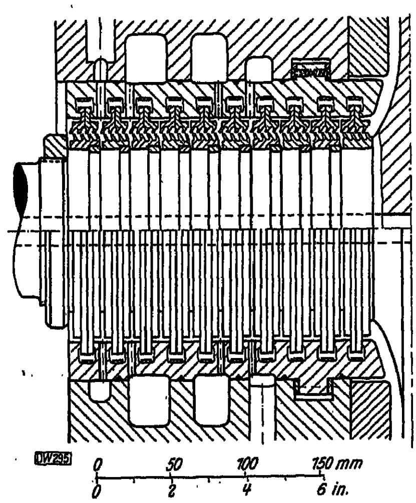


Fig. 66. *English Electric*, labyrinth-type packing gland for high pressures

the diaphragm gland used by the *Amer. G.E.C.* will be described here (Fig. 68). The packing ring is made of brass and is divided into four segments. A small shoulder in the diaphragm prevents the segments from moving inwards but allows a certain freedoom in an outwards direction. Each pair of segments is held in place in the diaphragm by means of a spring, and screws near the diaphragm joint prevent them from rotating.

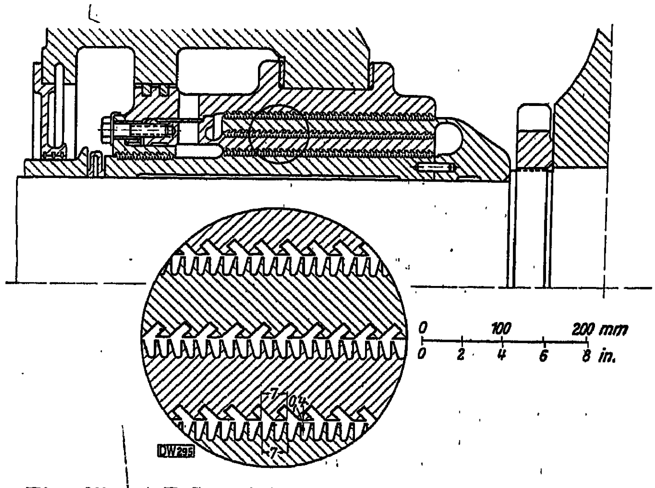


Fig. 67. *A.E.G.*, triple-flow labyrinth-type packing gland for high pressures

An intermediate method of tightening, between the type for disc turbines and the simple type for reaction turbines, is shown in Fig. 69. The *Brit. G.E.C.* use impulse blading on drums, as may be seen in Figs. 133 and 148. Thus, the intertage glands have almost the same diameter as the tips of the fixed blades. In order to provide an efficient labyrinth packing, however, a large coverband, in two rings, is fixed to the tips of the stationary blades. On the inside are the fixed labyrinth fins.

The steam escaping from the glands of back-pressure turbines is usually led away and used for feed-heating purposes. This arrangement can also be used for the steam from the glands of condensing turbines which is not used in the L.P. packings. If even a small whiff of steam escaping from the machine is not desired the outer part of the gland should be water-sealed (Fig. 70). These water-sealed glands consist, essentially, of discs rotating in annular spaces which are filled with water from the outside. The disc, which may be either smooth or provided with impellers, works in the same way as a centrifugal pump and produces a dynamic pressure. Thus, the outer pressure is balanced and causes the rotating water to have different levels, in a radial direction, on each side of the disc in the same way as a static pressure would affect an open fluid pressure gauge. Water-sealed glands are only used for turbines of large capacities as they absorb a considerable amount of power.

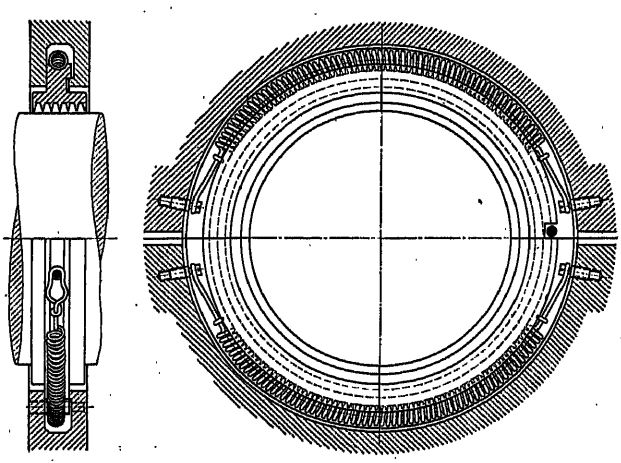


Fig. 68. *Amer. G.E.C.*, diaphragm packing gland

Formerly, carbon packings were much used for steam turbines, they were later almost completely supplanted by the labyrinth-type packings. At the present time, carbon packings are undoubtedly becoming more popular, especially in England and America. In Fig. 71 may be seen an H.P. outer gland of the carbon ring type of the *Amer. G.E.C.*. It contains four carbon rings in one box. Each ring is composed of four segments which are held together by a steel band. The free ends of this band are

joined together by a spring and a screw. Thus, when the shaft is cold or when the heat has made it expand, the carbon rings are always applied against the shaft with the same pressure. A leaf spring under each ring supports their weight and two stop pins are provided to prevent them from rotating with the shaft. In the lower part of the box are cast channels and flanges for connection to the pipes for leakage steam and drainage. The box is bolted to the turbine casing from outside and can be inspected without lifting the top part of the casing. Its cover is split vertically for this same reason.

As has already been mentioned, the glands of a condensing turbine are interconnected so that the leakage steam from the H.P. gland can be used for sealing the L.P. gland. When starting up it should be possible to supply throttled live steam to the glands, this, however, should only be done when the shaft is rotating. The diagram of the piping in Fig. 72 shows how the glands of a two-cylinder condensing turbine may be connected.

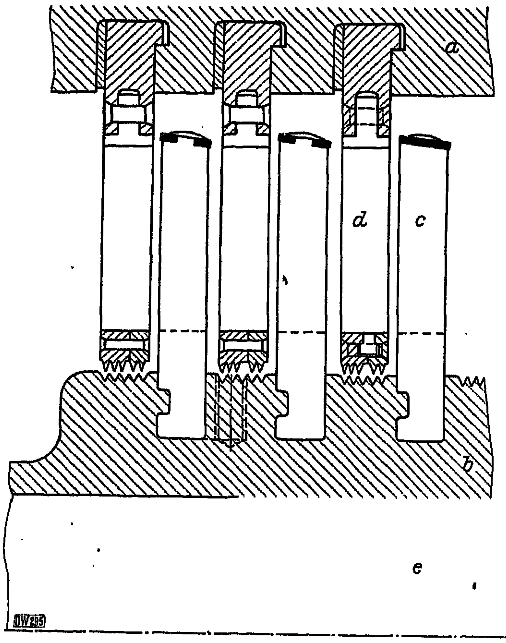


Fig. 69. *Brit. G.E.C.*, interstage labyrinth-type packing gland for H.P. impulse drums

a = Casing
b = Drum
c = Moving blade
d = Nozzle
e = Shaft

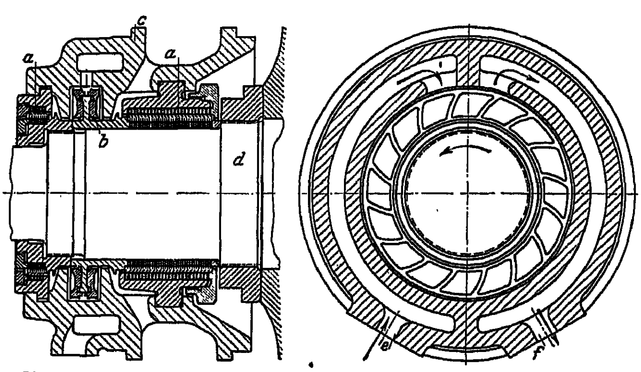


Fig. 70. *A.E.G.*, combined water-sealed and labyrinth-type packing gland

a = Labyrinth packing
b = Water seal
c = Housing of gland
d = Shaft
e = Water inlet
f = Water outlet

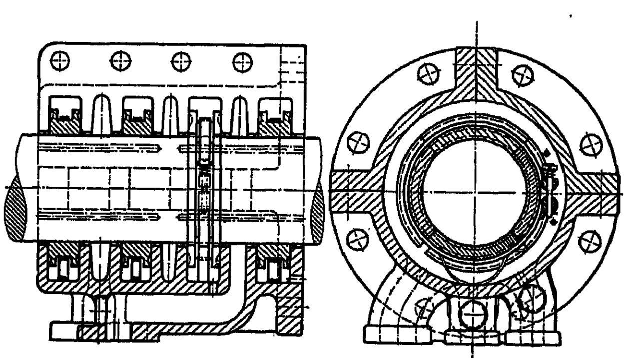


Fig. 71. *Amer. G.E.C.*, carbon ring-type packing gland

h. Bearings

The mechanical losses in steam turbines are caused by the thrust and journal bearings, the oil pump and the governor.

The thrust bearings are especially important. Formerly it was a fixed rule in turbine design that the thrust should be completely balanced or nearly so. Thrust blocks to take considerable loads were not

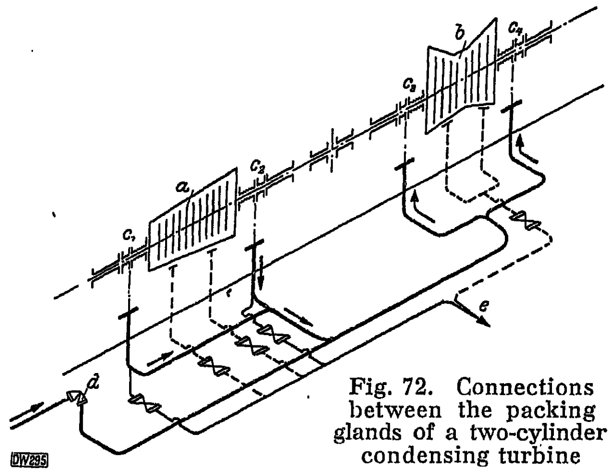


Fig. 72. Connections between the packing glands of a two-cylinder condensing turbine

Sealing steam from the H. P. into the L. P. packing glands and additional live steam
Leakage steam discharge
Drains from casings
a = H. P. turbine
b = L. P. turbine
c = Packing glands
c_1 = H. P. front end
c_2 = H. P. rear end
c_3 = L. P. front end
c_4 = L. P. rear end
d = Live steam throttle valve
e = To condenser

tolerated. This prejudice was quite justified in those days, for the multi-collar thrust bearing was universal and could only take small loads at the high turbine speeds. Thus, on account of the poor thrust bearing, reaction and combined reaction and impulse turbines had to be provided with a balance piston. This did not always lead, especially for single-casing machines, to either a simple or a reliable design. If the balance piston were eliminated, only stages for pure impulse or for a small reaction could be used in the case of single-flow turbines. It is obvious that the designing of turbines must have been greatly hindered by this limitation. Thus, for instance, a turbine such as that shown in Fig. 179, which is a simple, well-designed and efficient machine, could not be made unless the thrust block could be heavily loaded.

This difficulty no longer troubles the steam turbine builder. Single-collar thrust bearings are now almost exclusively used. They are capable of carrying considerable axial thrusts at high speeds, and with only a small friction loss.

The single-collar thrust bearing is an application of the *Reynolds* theory. It was brought out almost simultaneously by the Australian, *Michell,* and the American, *Kingsbury.* It can be explained by the recent theory on bearing surfaces (33), according to which the pressure in the liquid film between a stationary and a moving surface can only increase in the direction of flow (i. e. the direction of one of the moving surfaces) if the thickness of the film is variable. If the two surfaces, therefore, are parallel, they are unsuitable for taking any thrust, unless oil is supplied under pressure.

The invention of *Michell* and *Kingsbury* has had an enormous effect on the designing of both journal and thrust bearings. The principle of the invention is well known. The stationary bearing surface is divided into a number of segments which, under the influence of the rotation of the shaft and the load, are independently free to form a slight angle with the journal or collar (Fig 73). In this way the film between the collar and the stationary surface makes a number of wedges which are able to support very high pressures. According to theory, the friction coefficient is very small for low surface speeds and large loads, and this has been completely verified by tests. If the performances of *Michell* thrust bearings are compared with those of thrust bearings with parallel surfaces, it will be found that the friction coefficient has been reduced, by dividing the bearing surface and allowing the formation of wedges of oil, to about $^1/_{10}$ to $^1/_{20}$ of its former value, and is now about 0.003 to 0.0015.

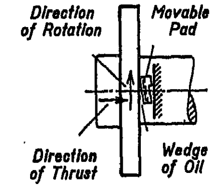


Fig. 73. Principle of the pad-type thrust bearing

(33) An exceptionally clear survey of the recent theory of bearings has been given by *Gümbel.* See the "Jahrbuch der Schiffbautechnischen Gesellschaft" 18 (1917) p. 236, and "Forschungsarbeiten," Issue 224 (Berlin: VdI.-Verlag 1920).

The form of the bearing surfaces is of great importance for the formation of a good film of oil. *Michell* had already discovered, by means of an apparatus for observing the lines of flow, that he bearing surfaces should not be too long in the direction of the flow. In a perpendicular direction they should, however, be wide, otherwise a large amount of the oil will escape over the edges.

Another important condition is that in the direction of the flow the pad should be supported on a pivot beyond the point of application of the resultant force of the oil pressure. Approximately a point at a distance of one-tenth of the width of the pad away from the centre can be taken as being the best position for the pivot. If a thrust bearing is to be suitable for both directions of rotation, the pads should be supported in their middle.

An average unit load of about 425 lb./sq. in. (30 kg./cm.²) may be assumed for a large thrust block. For small bearings it will be better to limit the maximum value at about 285 to 355 lb./sq. in. (20 to 25 kg./cm.²). The speed at the mean diameter of the pad may be as much as 213 ft./sec. (65 m./sec.). When either the velocity or the unit loading is increased, the amount of heat generated will be greater; therefore, if one of these two quantities is very high, the other one should be correspondingly reduced. It is usually advisable to raise the bearing pressure as near as possible to its limit value, this will enable smaller velocities to be chosen and the size of the bearing may be reduced.

Concerning the lubrication and cooling of single-collar thrust blocks, *Michell* has given a very useful chart (Fig. 74). For every velocity and for various bearing pressures the amount of oil required may be read directly for each cm.² of bearing surface and for a constant oil temperature rise of 18° F. (10° C.). The sections of the oil passages in the bearings should be ample, the oil going to the bearing should not flow faster than 4 ft./sec. (1.25 m./sec.) and in the drain it should not have a greater speed than 2.5 ft./sec. (0.75 m./sec.).

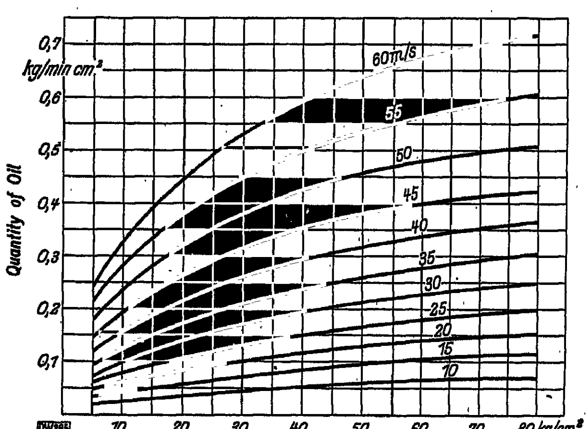


Fig. 74. Quantity of oil required in a pad-type thrust bearing per minute and per square centimeter of bearing surface for an 18° F. (10° C.) oil temperature rise, and for various surface speeds as a function of the bearing pressure

In the designing of single-collar thrust bearings of the pad type, two conditions have to be observed. Firstly, each pad should be able easily to take the proper slope; secondly, there should be no doubt about all the pads bearing equally.

The *A.E.G.* use in their steam turbines thrust bearings of *Michell's* design with pads having radial lines of support. The bearing shown in Fig. 75 is for a large thrust in one direction only; for a thrust in the other direction, only a flat bearing surface is provided. The collar supporting the pads is spherically seated, the collar on the shaft has a thin hub on one side and is held in position by a nut. It is important that the bearing surface of the collar should be exactly perpendicular to the shaft, a small inclination would prevent the pads all carrying equally. If the bearing has to be capable of taking thrusts in either direction, the portion of the bearing on the turbine side will still be built the same way and only the outer portion of the housing will be enlarged to contain another spherically seated ring and pads.

If the production is going to be extensive, the supporting rings, pads and patterns for housings may easily be made so that they can be largely used for types of different size. The oil is supplied in the centre, as is done in most designs, and is thrown outwards by the centrifugal force.

The *B.B.C.* thrust bearing fulfils the same requirements by resting the pads on balls, enabling them to turn, thus securing an even inclination of the pads and

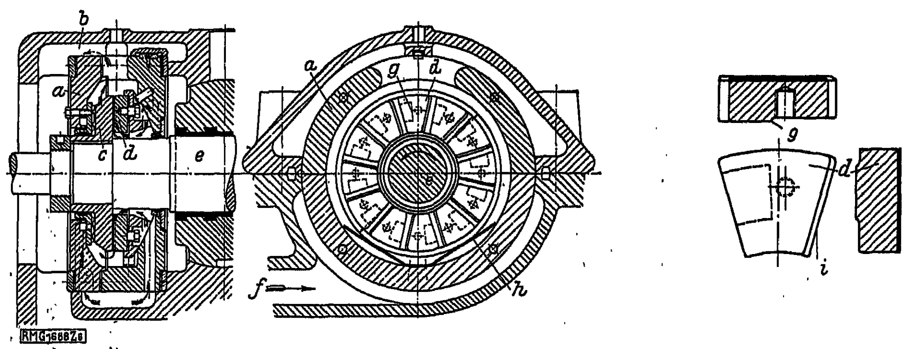


Fig. 75. *A.E.G.*, thrust bearing for thrusts in one direction with pads supported on a radial line. Scale about 1 : 14. Thrust 10 tons, speed 3000 R.P.M.

a = Bearing housing	*d* = Movable pad	*g* = Line of support
b = Oil drainage	*e* = Shaft	*h* = Supporting spring
c = Thrust collar	*f* = Oil inlet	*i* = Oil inlet edge of pad

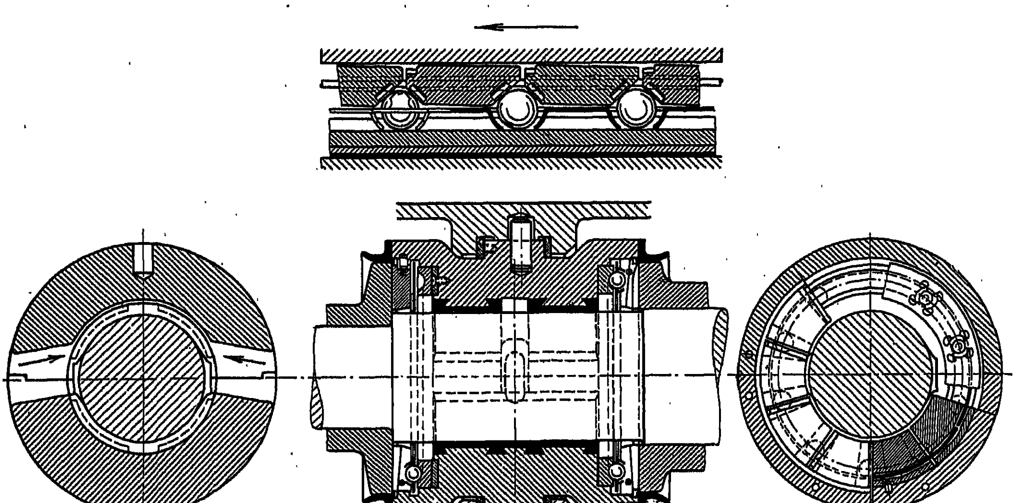


Fig. 76. *B.B.C.*, double thrust bearing with movable pads supported on points (balls), and journal bearing

an equal distribution of the load (Fig. 76). It can be seen from the sketch at the top of the illustration, if one of the pads gets heavily overloaded, the additional load will be immediately divided equally amongst the other pads. In the more heavily loaded bearings the housing is fitted with steel rings, and the pads with steel surfaces at the points of contact of the balls. In the type shown the two thrust bearings for the different directions are situated on either side of an ordinary bearing.

Westinghouse build thrust bearings of the *Kingsbury* type, which is commonly used in America. The pads are supported on points, as can be seen on the sketch on the right of Fig. 77. One of the supporting surfaces is flat, the other is spherical. They are both protected with hardened steel discs. Owing to their particular design, each pad automatically takes the same load as all the others. A special type of turbine thrust bearing is shown on the same figure which can take a large load for the diameter of its shaft. Two single-collar

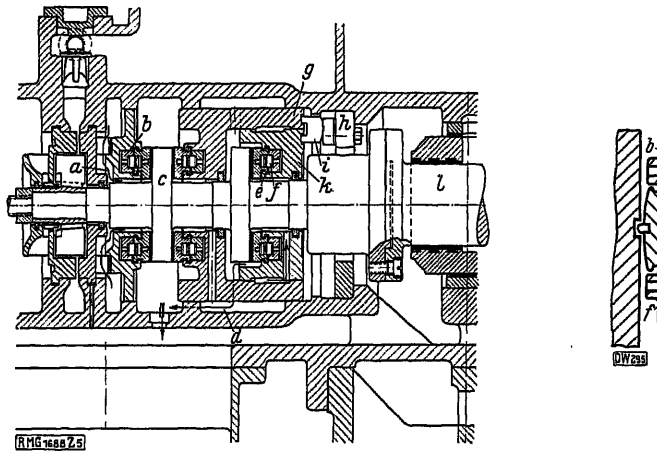


Fig. 77. *Westinghouse,* thrust bearing with two collars and movable pads supported on points (steel discs)

a = Oil inlet
b = Supporting rocker
c = Thrust collar
d = Oil drainage
e = Movable pad
f = Supporting disc
g = Housing of 1st thrust bearing
h = Supporting ring
i = Equalizing rocker
k = Housing of 2nd thrust bearing
l = Shaft

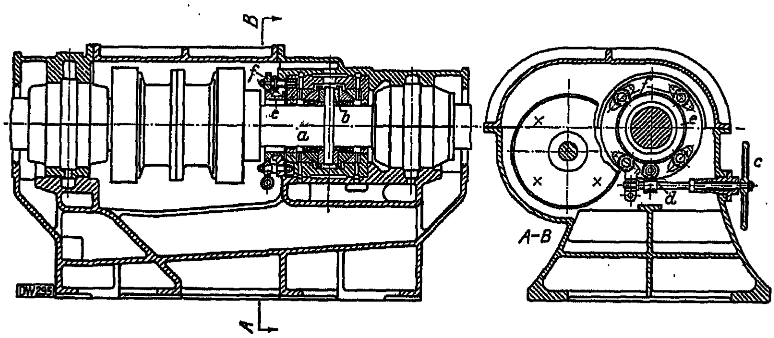


Fig. 78. *Parsons, Michell* thrust block with gear for adjusting the axial blade clearance

a = Shaft
b = *Michell* double thrust bearing
c = Hand-wheel
d = Worm
e = Intermediate wheel of adjusting gear
f = Adjusting screws with pinions

thrust bearings with pads have been placed one after the other in order to keep the diameter down and reduce the size of the shaft collar. In order that the two bearings may always be loaded equally, the one nearest the turbine is completely enclosed in the housing of the other. The two housings rest on the casing by means of a number of rockers which divide the load equally between the two bearings.

Parsons combine an adjusting gear with their single-collar *Michell* thrust block. Thus, the position of the rotor can be adjusted even when the turbine is running, and the small clearances may be obtained that are necessary in the

balance pistons and end tightened blading of *Parsons* turbines. In the example shown in Fig. 78, four screws are provided for this purpose in the bearing pedestal. They each have a pinion fixed to their heads, which may be turned from outside by means of a hand-wheel. By this means the whole thrust block is made to slide together with the rotor. *Westinghouse,* also, provide their single-collar thrust block with a worm and wheel for this same purpose, and other manufacturers use similar designs.

The principle of the *Michell* thrust bearing, or the division of the bearing surface amongst a series of separate adjustable pads for obtaining wedge-shaped films of oil, may also be used in journal bearings. Experimental bearings of this kind have been tried with complete success. In journal bearing, however, the method of working may be obtained approximately in a simpler and cheaper

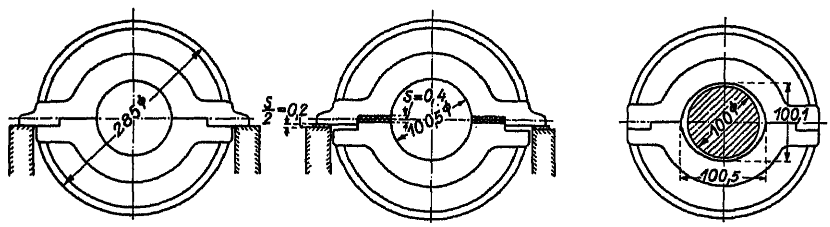
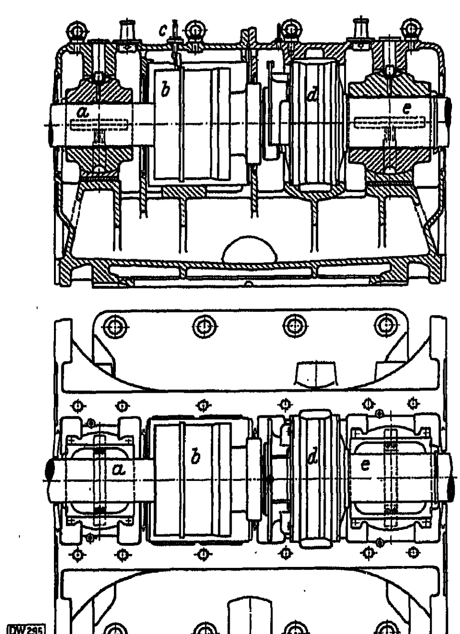


Fig. 79. *A.E.G.*, method of boring a turbine bearing bush

Machining the outer surface of the bearing bush — Boring the bush after insertion of plate — Bearing assembled and ready for use

way by boring the bush eccentrically. Fig. 79 shows the method of boring a bush of 3.937 in. (100 mm.) diameter. The halves have their outer surface machined together to the final dimension. They are then separated, a plate of 0.016 in. (0.4 mm.) thickness is placed between them, and they are bored out for a diameter of 3.957 in. (100.5 mm.). The plate is taken away and the bush is put in place. Thus, the clearance between the shaft and the bearing will be 0.020 in. (0.5 mm.) horizontally and only 0.004 in. (0.1 mm.) vertically. A wedge-shaped film of oil is formed which can carry the weight with safety.



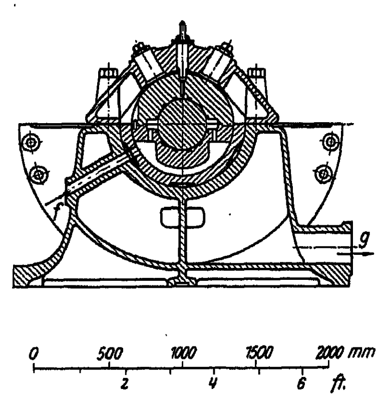


Fig. 80. *A.E.G.*, journal and thrust bearing in the centre pedestal of a two-cylinder condensing turbine

a = Rear journal bearing of the H.P. turbine
b = Flanged coupling
c = Clearance gauge
d = *Michell* thrust bearing
e = Front journal bearing of the L.P. turbine
f = Oil inlet
g = Oil drainage

In Fig. 80 a modern journal bearing for a steam turbine is illustrated. A bearing pressure of about 115 to 140 lb./sq. in. (8 to 10 kg./cm.^{2}) and a peripheral velocity of about 130 ft./sec. (40 m./sec.) may be allowed. The ratio of the length to the diameter of the bearing should be between 1.0 to 1.5. The amount of oil required can be calculated from the friction loss, the average value of the coefficient of friction may be taken as 0.008 for a bearing of good workmanship and for proper operating conditions. As regards the speed of flow of the oil, the remarks above concerning thrust bearings apply.

The bearings are usually supplied with oil by a system comprising oil pumps, coolers and pipes, which also is used for the relays and power pistons. The conditions to be observed when designing these important parts have been fully discussed in an earlier treatise of the author, to which only reference will be made here (34).

i. Rotors

Fig. 81 shows a variety of characteristic rotor forms. The rotors of impulse turbines with small diameters are made from solid forgings. If the speed of rotation is very high only a drum design is possible, even in the case

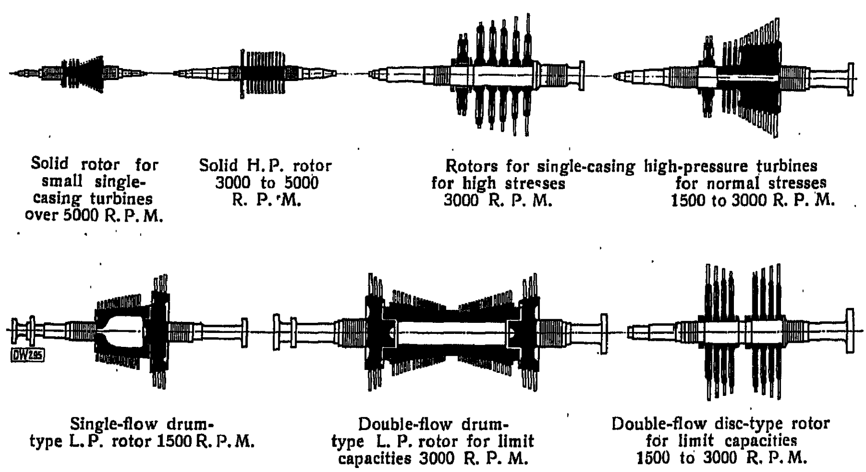


Fig. 81. *A.E.G.*, types of turbine rotors

of an impulse turbine (Type 1). For small speeds discs are cut out of the solid (Type 2). With larger diameters and higher stresses the discs are made separately in order that a sound forging may be secured. They are then shrunk on the shaft (Type 3). Small reaction drums are cut out of the solid, together often with the impulse discs (Type 4). Multi-casing turbines often have for the L.P. stages solid or hollow drums (Type 5). This is the simplest and strongest form of rotor. In order to increase the strength the last three or four stages are carried on a separate disc, out of which is also forged the exhaust portion of the shaft. The two forgings are shrunk and bolted together. In the case of double-flow L.P. turbines the drum is, naturally, joined at each end to a separate forging (Type 6). L.P. turbines with fewer stages are designed with disc-type rotors (Type 7). As regards the designing of discs and their mounting on the shaft, a distinction should be made between the H.P. discs and the L.P. discs. The H.P. wheels have smaller diameters, lower peripheral speed and are moderately stressed. They have to be designed

(34) Werft, Reederei, Hafen 4 (1923) p. 411.

especially to be as narrow as possible, so as to obtain the shortest machine and the smallest hub diameter and diaphragm glands. For this reason, instead of using, as formerly, separately mounted discs, the wheels are often made now in one piece with the shaft, like Type 2 of Fig. 81. This method, however, has also its drawbacks. For instance, flaws in the material may only be discovered after the machining is far advanced, or the rotor may get damaged and, even if only one stage is affected, the whole rotor may have to be replaced. This can be very inconvenient as it may take a considerable time to obtain a new forging.

The L.P. wheels have the highest peripheral velocities, and must be able to withstand high stresses. They must be made to decrease greatly in thickness from the hub to the rim; they must not have too large holes bored through them, and they must, especially, be mounted carefully on the shaft. These conditions apply also to the I.P. wheels of single-casing turbines, which are usually made with the same diameter. Discs of large diameters are not made in one piece with the shaft as large forgings cannot be manufactured sufficiently uniform, also the question of replacement of parts has to be

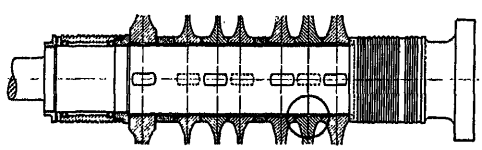
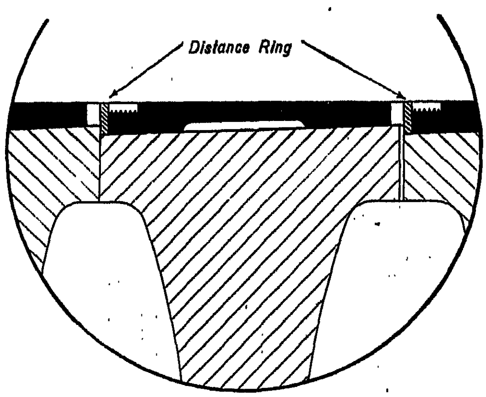


Fig. 82. *A.E.G.*, mounting of discs on conical sleeves



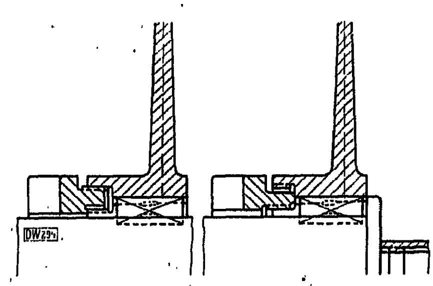


Fig. 83. *de Laval*, method of mounting and withdrawing discs

considered. Naturally, a wheel of given diameter may be made much lighter, especially at the hub, for a lower speed of rotation and the method of mounting on the shaft may be simpler.

A well-tried method is shown in Fig. 82. A conical, split and flexible sleeve is mounted on the shaft and over it a wheel is forced by hydraulic pressure. The wheels are all held in position by a single nut. The withdrawing of the wheels is very simple and can be performed in a manner similar to that employed for the *de Laval* type (Fig. 83). When the wheel is being mounted a nut is used which fits into a screw thread on the sleeve and pushes the hub. When the wheel is being dismounted, the nut screws into the hub, and pushes against the sleeve.

For mounting either the discs of single-casing turbines, or the L.P. discs and the drum-type discs for carrying several H.P. stages in multi-casing turbines, *B.B.C.* use U-shaped flexible rings which press against the shaft and the hub (Fig. 84). Owing to their U-shaped section the rings deflect easily under a concentric pressure; they are, however, very rigid against an eccentric pressure. Thus, the hub may expand under the centrifugal force or the temperature without the centering being affected and without the wheels coming loose.

It is important for machines with many stages and for high temperatures that the hubs should not be too thick or too heavy so as not to lengthen the machine. The wheels, also, should not come loose under the influence of the varying temperature. The method of the *Amer. G.E.C.* is good from these points of view. Between the hub and the shaft is a cylindrical sleeve which is connected to the hub by a number of radial pins and is keyed to the shaft. The wheel is shrunk on to the shaft, together with the sleeve, in the usual way. If there is a sudden change in temperature, such as might occur when starting

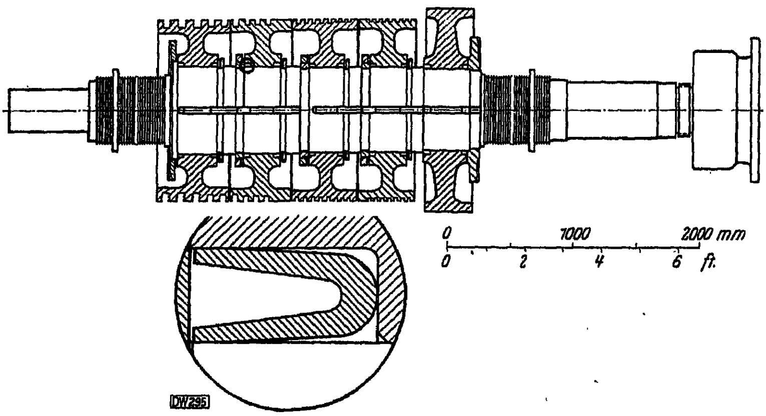


Fig. 84. *B.B.C.*, 1800 R.P.M. H.P. rotor of a 160,000 kw., two-casing, high-pressure turbine, and method of mounting discs on shafts by flexible rings of U-section

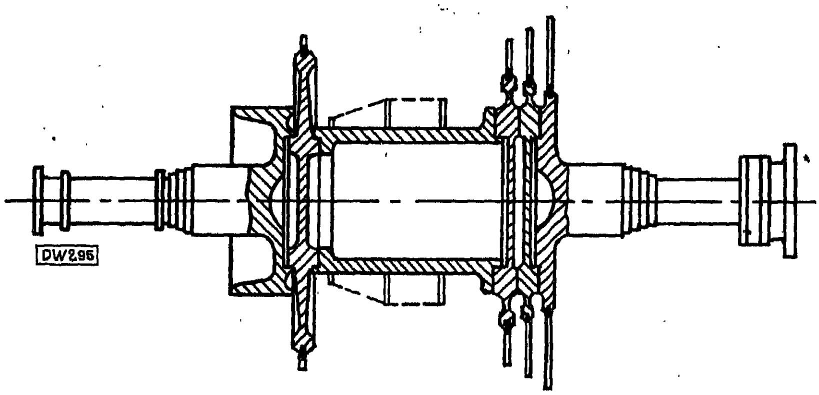


Fig. 85. *S.S.W.*, rotor of a single-casing high-pressure turbine of the *S.S.W.-Roeder* type

up, the wheels get hot quicker than the thick shaft, and they may expand independently of the shaft. They separate from the sleeve, but the pins keep them centered and transmit the turning moment.

As an example of a rotor of the *Parsons-B.B.C.* pure reaction type, the H.P. rotor is shown of the well-known Hell Gate turbine of 160,000 kw. (Fig. 84). The usual solid drum has been abandoned on account of the large diameter and load, and to economize weight and facilitate heat expansions. The stages are arranged in groups on wide discs all of the same diameter. They are each mounted on the shaft by means of flexible rings, as has just been described. The balance piston is made in a separate disc.

It is not necessary to mention other types of rotors here, when describing examples of actual turbines (from page 101 onwards) all the more important types will be found. On account of several original features, reference is made, however, to the rotors of the *Westinghouse* turbines, such as the one shown in Fig. 146, and to the rotors of the *S.S.W.-Roeder* turbine in Fig. 85, which is a development of the *de Laval* type.

Closely connected with the rotors are the couplings. Their usual designs, such as the rigid flange type, or the numerous claw or tooth types, are so well known that they need not be discussed here in detail. An excellent form of double toothed coupling is described later in connection with gear drives (page 172). We shall only mention here a special coupling of recent design and of the rigid type, which is, however, flexible to some extent (Fig. 86). The halves are connected by a corrugated pipe which allows a certain flexibility. More complicated types, such as the *Bibby* or the *Schürmann* couplings, will only be mentioned here as they have not yet been able to gain a wide application for steam turbines.

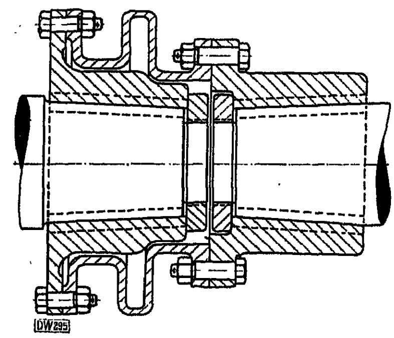


Fig. 86.
Metro-Vick, coupling of the corrugated pipe type

j. Governing

The governing of steam turbines has always been considered a special problem which can only be understood by specialists. The question is, however, coming more and more to the fore as the requirements become more stringent. As examples, only the extraction and mixed-pressure industrial turbines, turbines working with steam from accumulators, and primary turbines for very high initial pressures, need be mentioned here. As the importance of governing is growing from year to year in steam turbine construction, it may be considered as being within the scope of this book. It can only be treated, however, under the assumption that the various parts of governor gears are known. They have been comprehensively and clearly described by *Stein* (35) and *Tolle* (36). When a survey is attempted of the most modern methods of governing steam turbines, it can only be done by choosing a few characteristic examples from amongst the numerous solutions.

Two principles are common to all governor gears: The speed governors regulate the load, and the pressure governors regulate the steam quantities. Turbines which only have to govern their load, such as high-pressure turbines especially, will, for this reason, only have a speed governor. When speed and pressure governors work on the same regulating elements they may sometimes put each other out of action at the end of their movement. If they work separately on the regulating elements the operation of one governor usually causes the other governor to actuate its own regulating element because any change in load upsets the balance of the steam quantities and, conversely, any change in the steam flow alters the output. When any condition changes, therefore, both the speed and pressure governors must operate in order to re-establish equilibrium. For this reason, combined governing, as introduced by *Rateau*, is used as much as it is adaptable to the different systems. *A.E.G.*, *B.B.C.* and *S.S.W.*, for instance, employ such a method. Each governor acts on all the regulating elements simultaneously in such a way as not to affect the equilibrium of the quantities which do not require governing at that moment.

(35) *Th. Stein:* "Regelung und Ausgleich in Dampfanlagen" (Berlin: J. Springer 1926).
(36) *M. Tolle:* "Regelung der Kraftmaschinen", 3rd Edition (Berlin: J. Springer 1921).

The governor gear of the *A.E.G.* turbines is of the lever type. The movement of the speed or pressure governors is transmitted by articulated levers to the relays. The smaller sizes of throttle valves are single-seated, the larger sizes are double-seated. They are operated singly or in groups by a camshaft which is turned by a rotating power piston. The throttle valves of large high-pressure turbines are usually placed at the side of the machine, those for small turbines or industrial turbines are generally on the top of the turbine casing. The governor gear of a high-pressure turbine, which is a plain speed regulator, is part of the gear of every special turbine. For this reason, and because the *A.E.G.* have had a wide experience in special turbines, two characteristic *A.E.G.* governor gears will be briefly described. One is for a turbine working with live steam and steam from an accumulator, and the other is for a double-extraction condensing turbine.

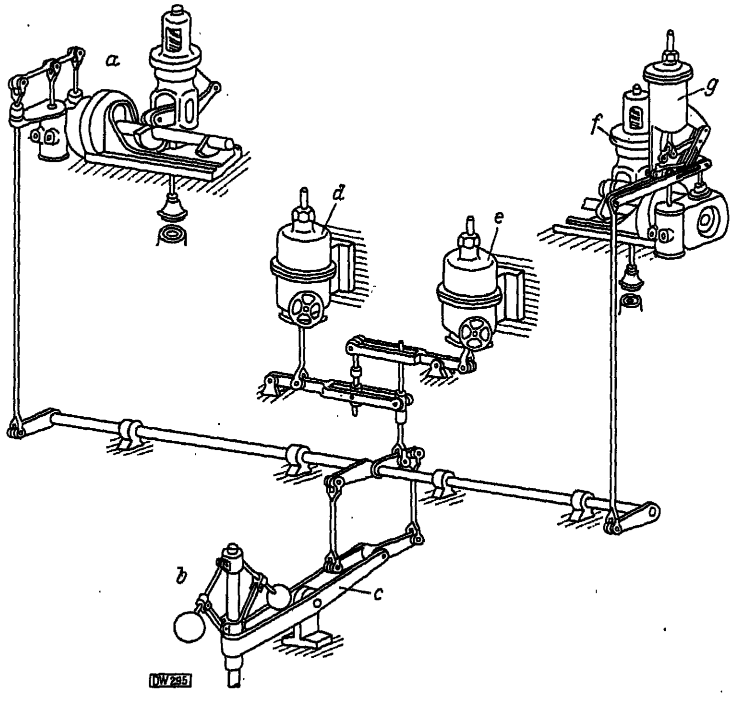


Fig. 87. *A.E.G.*, diagrammatic sketch of the governing arrangement of a combined live steam and accumulator steam turbine

a = Live steam control gear
b = Speed governor
c = Governor lever
d = Live steam pressure regulator
e = Pressure limiting regulator
f = Accumulator steam control gear
g = Accumulator steam pressure regulator

The diagram of a turbine for live steam and accumulator steam is shown in Fig. 87. When the load increases the speed governor *b* can open both the live steam valve *a* and the accumulator steam valve *f* by two independent sets of bell cranks. As long as the boiler pressure is constant, however, the live steam pressure governor *d* will prevent the opening of the accumulator steam valve. If the boiler pressure falls, the live steam valve will be closed and low pressure steam will be admitted to the turbine. When the pressure in the accumulator becomes about equal to the pressure in the first stage, the pressure limiting regulator *e* will close the low pressure valve and prevent steam from the turbine returning to the accumulator. A third pressure regulator *g* is provided to prevent a decrease in the load through a falling off in the accumulator pressure after the set has run a long time on mixed-pressure. It enables the

accumulator to be charged at a rate depending on its pressure. To explain this, let it be supposed that there is no accumulator pressure governor. Several sets may be running in parallel, those not constructed for using accumulator steam might get overloaded if the load of the sets designed for mixed-pressure were to fall off. The boiler plant would also be overloaded, the live steam pressure would fall still further and the function of the steam accumulator, which is the maintaining of a constant boiler pressure, would not be fulfilled.

The method of governing *A.E.G.* turbines will be clear from this diagram. Each movement of a governor not only adjusts one gear, but alters simultaneously all the other ones to suit.

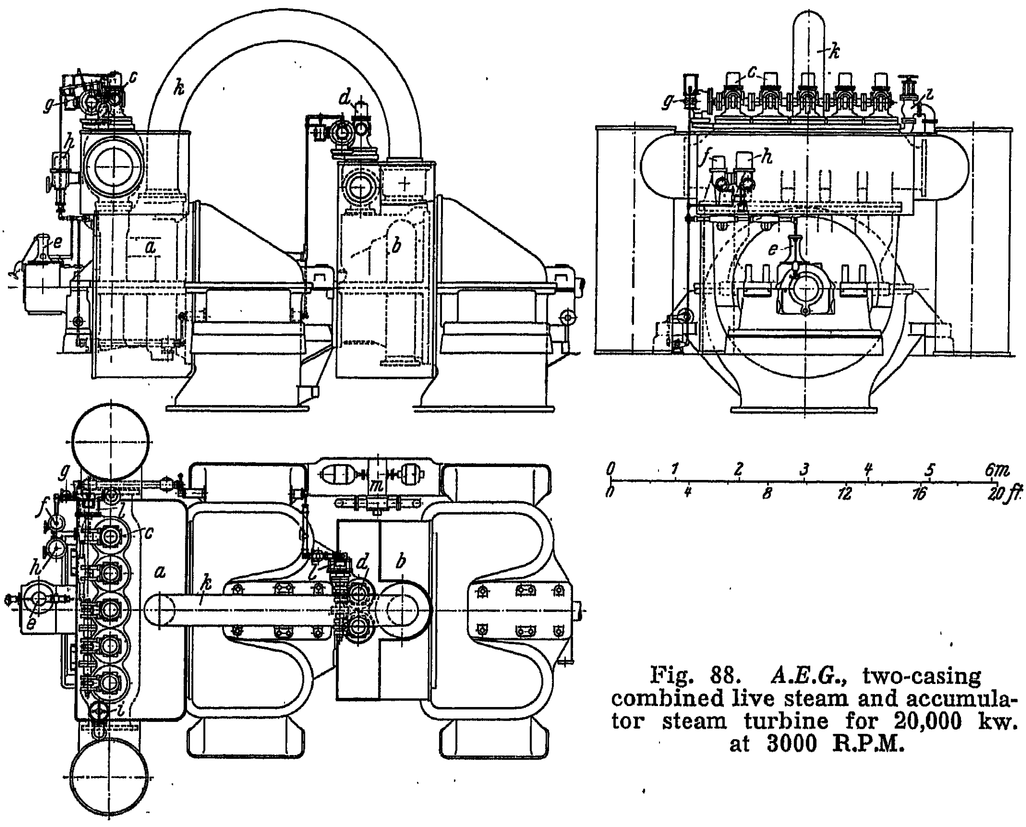


Fig. 88. *A.E.G.*, two-casing combined live steam and accumulator steam turbine for 20,000 kw. at 3000 R.P.M.

a = Accumulator steam turbine
b = Combined live and accumulator steam turbine
c = Accumulator steam control gear
d = Live steam control gear
e = Speed governor
f = Boiler pressure limiting regulator
g = Pressure correcting regulator
h = Pressure difference regulator
i = Air valve
k = Overflow connection for accumulator steam
l = Rotating power piston
m = Emergency oil pump driven by A. C. and D. C. motors

Fig. 88 shows a governing arrangement very similar to the one just described. The turbine is for a station to take peak loads, and is probably the largest machine ever built for running with live and accumulator steam. The steam conditions are 285 lb./sq. in. (20 kg./cm.²) gauge, 700° F. (375° C.) and the cooling water temperature is 59° F. (15° C.). The accumulator steam pressure fluctuates between 185 and 7 lb./sq. in. (13 and 0.5 kg./cm.²) gauge. The turbine is in two casings and can give a maximum output of 20,000 kw. at 3000 R.P.M. when operating with only live steam or with only accumulator steam at a pressure not less than 71 lb./sq. in. (5 kg./cm.²) gauge. The largest amount of accumulator steam which may flow through the turbine is no less than 206 tons

per hour. The first casing only receives accumulator steam, the second one works with only live steam in its first stage, the other stages work mixed. When running only with live steam at a normal pressure, the regulator for limiting the boiler pressure acts in such a way as to prevent the speed governor being able to open the accumulator steam valves. If the live steam pressure falls below normal, the boiler pressure limiting governor releases the accumulator steam valves and the turbine begins to run on mixed-pressure. The pressure in the accumulator falls gradually if steam is drawn off for a long time. The load can only be maintained by means of a regulator which opens the L.P. valves as the pressure in the accumulator sinks. The movement of this pressure regulator is transmitted to the relay by means of a suitably arranged

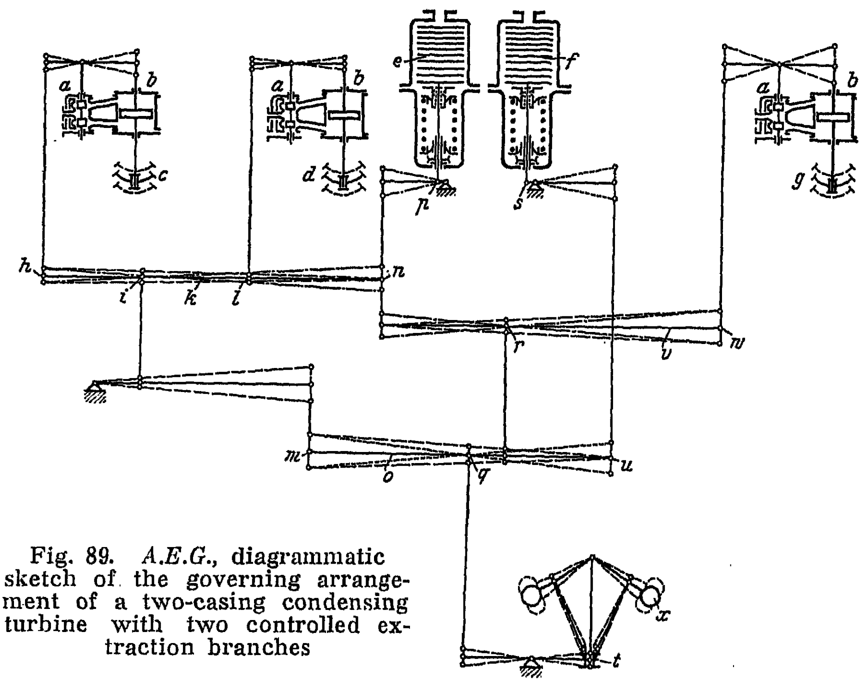


Fig. 89. *A.E.G.*, diagrammatic sketch of the governing arrangement of a two-casing condensing turbine with two controlled extraction branches

a = Pilot valve
b = Power piston
c = First extraction regulating valve
d = Live steam throttle valve
e = First extraction pressure regulator
f = Second extraction pressure regulator
g = Second extraction regulating valve
$h, i, l, m, n, p, q, r, s, u, w$ = Pivots
k, o, v = Levers
t = Governor sleeve
x = Speed governor

gear. This adjusts the position of the valve according to the charge of the accumulator and the load is maintained approximately constant without the intervention of any other governor. As soon as the boiler pressure has fallen a certain amount below its normal value, the boiler pressure limiting governor completely closes the live steam valves and admits enough accumulator steam to hold the load without the intervention of the speed governor. The pressure difference regulator *h* prevents steam flowing into the accumulator from the first turbine stage when the two pressures are about equal. It closes the accumulator steam valves and, at the same time, it counteracts the boiler pressure limiting regulator and opens the live steam valves. In this way the set can be kept running with live steam at a pressure below the normal. The boiler pressure limiting governor can be set for pressures between 285 and 185 lb./sq. in. (20 to 13 kg./cm.^{2}) gauge.

The second example is a two-cylinder condensing turbine with two controlled extraction points. The diagram is shown in Fig. 89. The load is regu-

lated by a speed governor which acts on the H.P., I.P. and L.P. regulating valves in such a way that neither extraction quantity is affected by a change in the load. Each extraction branch has a governor for maintaining the pressure constant. If less steam is required in the first branch the pressure will rise slightly. This causes the pressure governor to close the live steam valves and to open the regulating valves before the I.P. and L.P. parts. Less steam will then be admitted to the turbine through the live steam valves, whilst the I.P. and L.P. valves will let more steam through. In this way the extraction quantity will diminish. The leverages are chosen so that the closing of the H.P. valves and the opening of the I.P. and L.P. valves do not influence either the load or the second extraction quantity. The pressure at the second branch is kept constant in a similar manner. Thus, if less steam is required at this point, the pressure governor will close the live steam and the first extraction valves and open the second extraction valves. The load and the quantity of the first extraction are not affected by this operation. Each pressure governor is provided with a gear which enables it to be put out of action if required. Hence, the turbine may work as a machine of any of the following types: Firstly, as a plain high-pressure turbine; secondly, as an extraction turbine with one branch which may be for either a higher or a lower pressure; thirdly, as a double-extraction turbine; fourthly, as an extraction back-pressure turbine with the L.P. turbine disconnected.

S.S.W. use a similar principle for their governing. The power pistons are, however, placed near the governors. In Fig. 90 it may be noticed in particular how the emergency and safety devices are connected with the governing arrangements. The pressure of the governor oil is applied, through the pipe n_3, under a piston on the emergency valve *b* pressing it against an oil release valve *m*

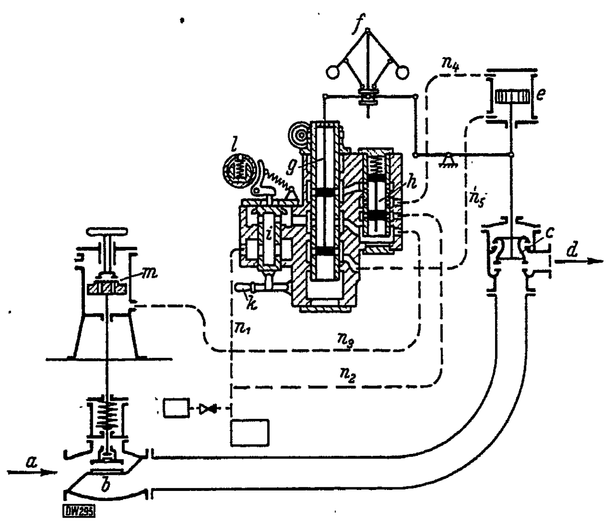


Fig. 90. *S.S.W.*, diagrammatic sketch of the governing and emergency arrangement of a high-pressure turbine

a = Live steam
b = Stop and emergency valve
c = Throttle valve
d = To turbine
e = Power piston
f = Speed governor
g = Pilot valve
h = Relay piston
i = Emergency trip bush
k = Hand-operated tripping lever
l = Emergency governor
m = Power piston and oil release valve
$n_{1 \div 5}$ = Oil pipes

and maintaining the emergency valve open whilst the set is running. When starting up the emergency valve may only be opened slowly by raising the oil release valve with a hand-wheel. If the hand-wheel is turned too quickly, the piston cannot follow fast enough, the oil below the piston is let out at the top, the pressure falls and the emergency valve is closed by a spring. Thus, the turbine cannot be damaged by a sudden opening of the valve. At the same time it is impossible to start the set unless the oil pump is delivering enough oil, for the piston will only follow the movement of the release valve when the oil pressure is sufficient. An unexpected drop in the oil pressure during operation, from a failure of the oil pumps or a leak in the piping, will trip the emergency valve in the same way as the emergency governor. In the latter case, however, the bush *i* and the piston *h* move

downwards under their own weight. Tripping always takes place without the use of any intermediate levers.

When possible, the throttle valves are placed on the top of the H.P. casing of the turbine. The valves are only arranged in front or at the side of the machine when the governing is exceptionally complicated. Fig. 91 shows a section through the governor drive and the power piston of the cam shaft for a high-pressure turbine. The oil pressure rotates the piston in either direction. The valves are opened positively. They are designed to close with springs only, if they should stick, however, cams and rollers are arranged to force them down.

The stability of the governing is increased by an indirect compensating movement by means of a disc having a flat spiral slot.

Escher Wyss use a gear for their throttle governing which differs somewhat from those already described. An original method

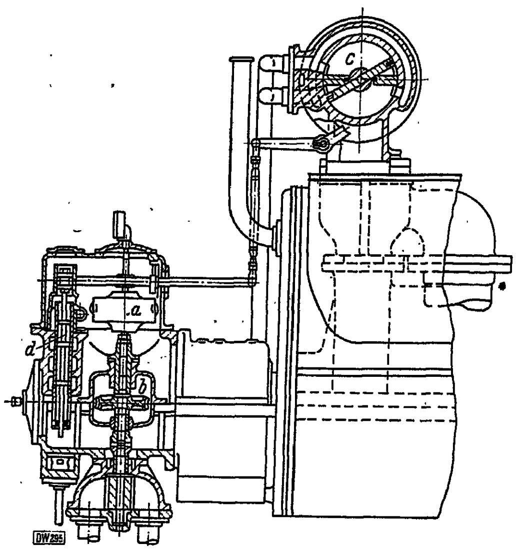


Fig. 91. *S.S.W.*, governor drive and power piston of a high-pressure turbine

a = Speed governor
b = Governor worm drive
c = Rotating power piston of the cam shaft
d = Relay valve

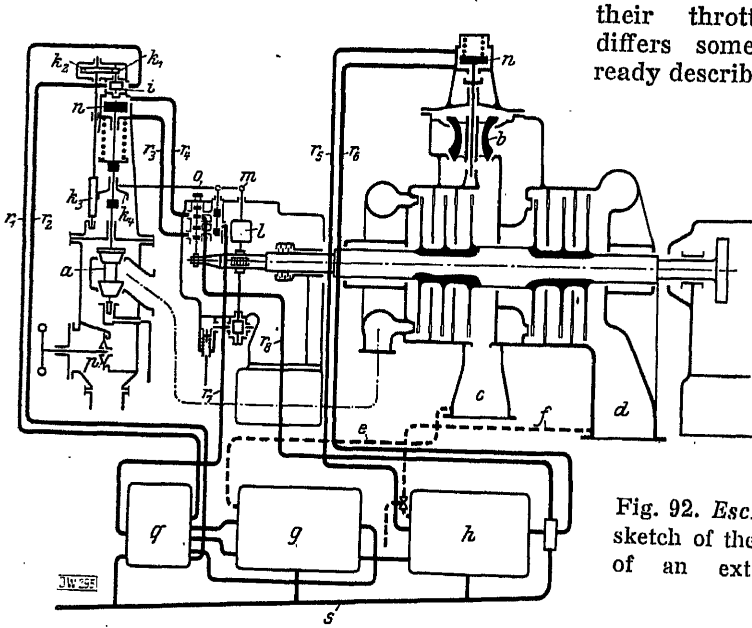


Fig. 92. *Escher Wyss*, diagrammatic sketch of the governing arrangement of an extraction back-pressure turbine

a = Live steam throttle valve
b = Extraction regulating valve
c = Extraction Branch
d = Exhaust branch
e = Steam pipe to extraction pressure governor
f = Steam pipe to back-pressure governor
g = Extraction pressure governor
h = Back-pressure governor
i = Oil motor

$k_{1 \div 4}$ = Toothed wheels
l = Speed governor
m = Pivot
n = Power piston
o = Governor lever
p = Stop valve
q = Speed adjusting gear
$r_{1 \div 8}$ = Oil pipes
s = Oil drain pipe

is employed for the governing of turbines with more than one gear. All the regulating valves do not move together and in co-ordination to a position of equilibrium, they are operated independently. The principle will be explained with an example of an extraction back-pressure turbine (Fig. 92).

If, for instance, the extraction pressure rises, the extraction pressure regulator will rotate the oil motor *i*, which in turn rotates the toothed wheel k_4 through the toothed wheels k_1, k_2 and k_3. It may be mentioned that the *Escher Wyss* pressure governor works entirely without levers, further details on this point cannot be given here. The wheel k_4 is made as a nut and, in this case, it is screwed on to the spindle of the main throttle valve. The point *m* is fixed as the position of the speed governor does not vary. The movement upwards of the wheel k_4 will, therefore, raise the pilot valve, oil is led to the top of the power piston and the valve closes. The compensating movement is performed by the lever *o*. The relay *i* continues to adjust the main throttle valve until the pressure in the extraction branch is normal again. The position of the speed governor is only affected by the frequency of the current in the entire system of power production. The regulation of the back-pressure is done in a similar way. The back-pressure governor comes into action and adjusts the extraction valve. This causes a fluctuation in the extraction pressure, which in turn affects the regulation of the live steam, and the throttle valve is adjusted to the new state of equilibrium. This system of governing is not compound like those of the *A.E.G.* and *B.B.C.*.

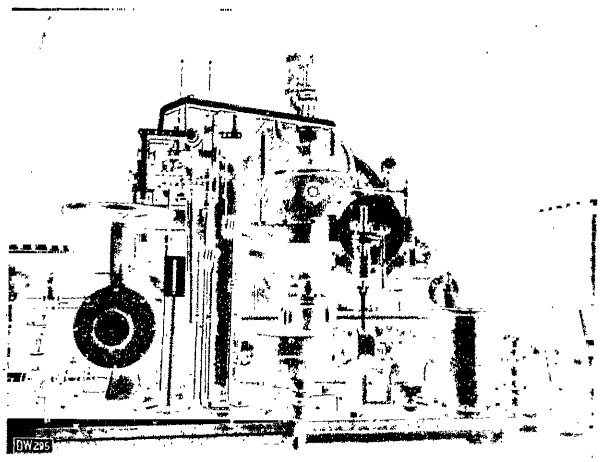


Fig. 93. *Allis-Chalmers*, front view of a 5000 kw., 3600 R.P.M. extraction turbine on the test bed

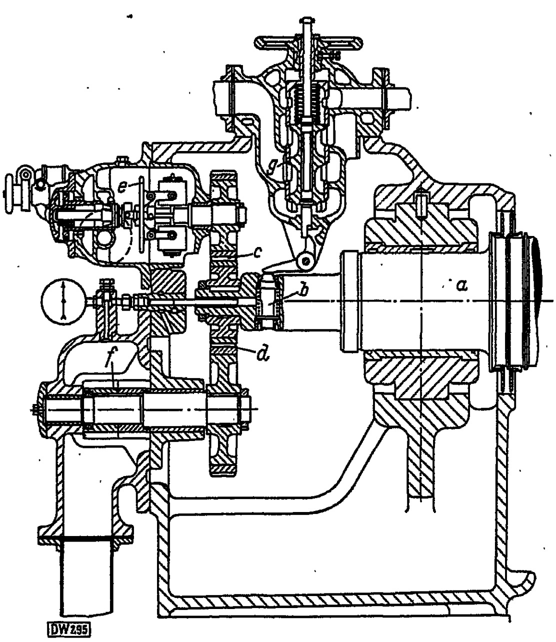


Fig. 94. *B.B.C.*, section through the governor, the operating gear of the stop valve and the emergency tripping gear

a = Turbine shaft
b = Emergency governor
c = Spur wheel driving governor
d = Spur wheel driving oil pump
e = Speed governor
f = Oil pump
g = Starting gear

Methods of governing, similar to the three which have just been described, are used by a large number of other firms. *Allis-Chalmers*, for instance, govern their emergency valve by means of a power piston, and they operate their throttle valve by

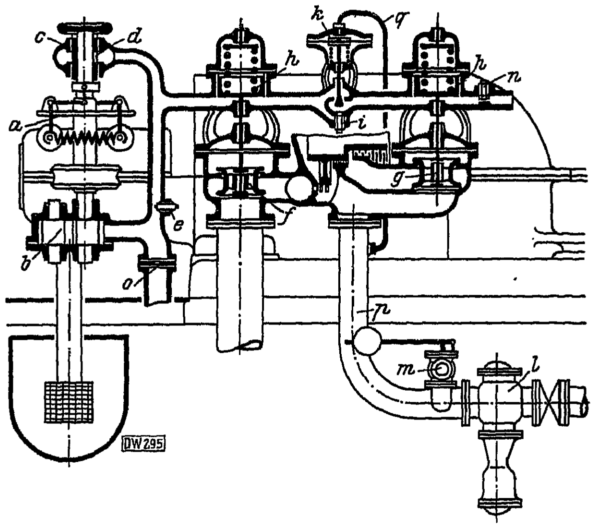


Fig. 95. *B.B.C.*, diagrammatic sketch of the governing arrangement of an extraction turbine

a = Speed governor
b = Gear-type oil pump
c = Bush controlling the oil pressure
d = Oil release slot
e = Valve for adjusting oil pressure
f = Live steam throttle valve
g = Extraction valve
h = Power piston
i = Disconnecting valve
k = Diaphragm-type pressure governor
l = Non-return valve
m = Relief valve
n = Valve for adjusting the oil drainage
o = Throttle diaphragm
p = Extraction branch
q = Steam pipe to pressure governor

a lever which is connected to the spindle of the emergency valve. Only in the case of extraction turbines are the valves of the extraction steam operated by separate power pistons. Fig. 93 shows very well how complicated may appear the collection of valves, rods, gauges and other parts belonging to a governing arrangement. It is a picture taken in *Allis-Chalmers'* Works.

The governing employed by *B.B.C.* differs from that already described by being of a continuous oil flow type without any levers for the compensating movement. A flow of oil at a constant initial pressure is continuously sent through the main oil pipe of the governing system, to which are connected the power pistons of the valves. These are suitably loaded with springs. A branch of the oil main goes to the speed governor and is joined to an annular space around the governor sleeve. The movement of the sleeve uncovers to a certain extent a slot which allows a variable amount of oil to escape from the annular space. When the load rises the governor lifts the sleeve and reduces the size of the aperture. The pressure then rises in the oil main and under the power pistons of the throttle valves. A new position of equilibrium is arrived at between the force of the spring on the top of the piston and the pressure of the oil underneath. In this way, to every position of the governor corresponds a certain output.

Fig. 94 shows a section through the governor gear. The centrifugal governor is on a separate spindle, which is driven by the main shaft through spur wheels in the same way as the gear-type oil pump. In this governing arrangement, not only the throttle valves, but the stop valve, also, is operated by oil. Thus, starting up is facilitated and can be performed with a single hand-wheel. It first closes the throttle valves, then it successively opens the stop valve by-pass, the stop valve and,

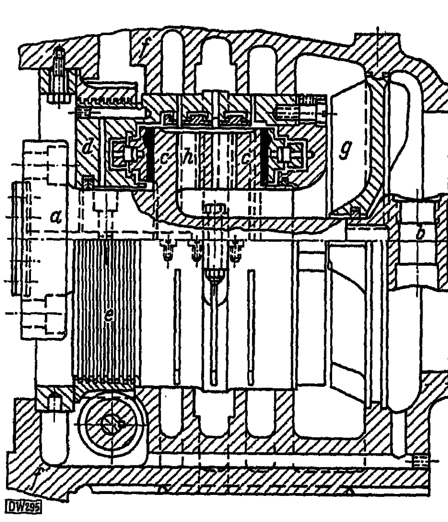


Fig. 96. *Westinghouse*, impeller of the oil governor and *Kingsbury* thrust block of a large high-pressure turbine

a = Turbine extension shaft
b = Emergency governor
c = Thrust collar
d = Housing of thrust bearing
e = Rotor adjusting gear
f = Pedestal
g = Oil suction
h = Oil pump

finally, the throttle valves are again opened. The same operations take place in the reverse sequence when shutting down or when the machine is tripped, as the emergency governor acts on the relay valve of the starting gear. For very large machines an intermediate relay is used between the governor sleeve and the power pistons.

Turbines which are governed by the steam flow, instead of the load, have pressure regulators of the diaphragm type which are connected with valves for throttling the oil flow. As an example the case of a back-pressure turbine coupled to a large electrical system may be mentioned. When the back-pressure falls, the pressure regulator if it is placed after the power pistons of the throttle valves will close the oil valve in the drain. Thus, the pressure under the power pistons is increased, and the valves open wider. The sleeve of the speed governor is adjusted so that changes in load do not affect the oil pressure.

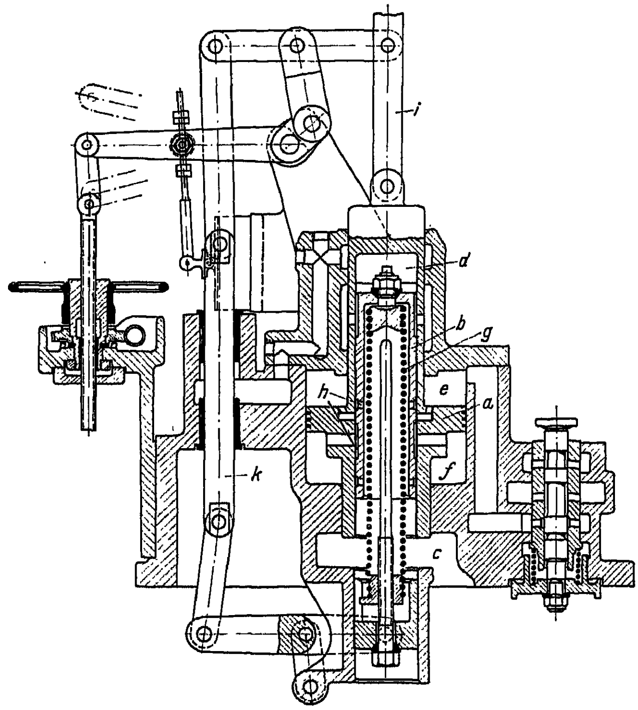


Fig. 97. *Westinghouse*, section through the relay of an oil pressure governor

a = Relay piston
b = Relay valve
c = Oil pressure chamber
d = Dash pot
e = Relay cylinder
f = Relay cylinder
g = Spring
h = Equalizing ports
i = Connecting link
k = Follow-up rod

In the case of extraction turbines, the pressure governor and its oil throttling device are connected in the oil circuit between the throttle and the extraction valves. In this way the oil main is divided into H.P. and L.P. portions (Fig. 95). When the load increases, the change in speed causes a variation of the oil pressure in the same direction in both systems. Thus, both the throttle and the extraction valves will open. If, for instance, the extraction pressure falls at the same time, the pressure regulator will throttle the oil between the throttle and extraction valves. The power piston on the extraction valve will receive less oil and the valve will close. Turbines with more than one pressure regulator and more than two governing systems may be controlled on this same principle.

Westinghouse and the *Brit. G.E.C.* use a governing arrangement which, although it is not without levers, is different from those already described by the design and operation of the governor. The operating force is derived from a centrifugal oil pump which is coupled to the main shaft. The oil pressure varies as the square of the turbine speed. Fluctuations in the oil pressure are directly used to operate the relay.

Fig. 96 shows the governor of a 60,000 kw. *Westinghouse* turbine. The impeller of the oil pump has been machined out of the collar of a *Kingsbury* thrust bearing. The oil is then led to the point *c* (Fig. 97) in the first relay and presses upwards against the relay piston *a*. In this piston a relay valve *b*, whose position is principally determined by the spring *g*, can move with a slight

play. The space *d* above the valve is connected to the main lubricating system and it is, thus, at the same pressure as the lubricating oil in the whole set. If the governor oil pressure rises at *c*, the relay valve *b* is raised. This reduces the section of the equalizing ports between *e* and *f*, which have been drilled in the relay valve and piston. The oil pressure at *e* then falls below that at *f*, the relay piston rises and operates the throttle valves by means of the link *i*. The relay valve is now brought back to its normal position in the piston by the rod *k*. Thus, the pressures at *e* and *f* are balanced and the piston stops.

The relay valve eliminates the possibility of a failure of the governing. If the throttle valves were to remain stuck open, the movement of the relay valve would not alter the position of the piston. The valve would, therefore, continue to rise until the spaces above and below the piston were put into communication by means of a slot. In this way, the whole pressure of the oil at *c* and the force of the spring strive to close the throttle valves.

The speed of the turbine depends on the tension of the spring *g* and on the time required to bring the piston valve back to its normal position in the relay piston. Thus, the running speed can, naturally, be accurately set by altering the tension of the spring *g*. However, the change is usually made by adjusting the levers of the compensating movement.

The governing movement is communicated from the relay piston to the valves by means of the usual power pistons, the follow-up movement being performed by levers.

II. Technical factors limiting the design

Any progress in design or arrangement has its application limited by the physical properties of the materials available at the time. As turbines are improved and the limits of construction approached it becomes all the more important that the designer should know the boundaries. In modern steam turbine construction, therefore, a knowledge of the materials has come to be of enormous importance. Hence, in a book which is especially devoted to recent improvements, the question of the present day limits must be treated.

1. Properties of the materials

a. Cast iron

Of all the substances contained in cast iron which determine its nature, carbon has the greatest influence. It is not so much the actual carbon content or the ratio of the elementary (graphite) to combined carbon (ferro-carbide) that matters, it is more the size, form and distribution of the elements of carbon.

Fig. 98. Microphotographs of cast iron. Magnified 100 times

These break up the continuity of the metallic matrix, and can be compared to enclosures of slag. The importance of the graphite can be understood if it is realized that one per cent in weight of graphite in iron occupies 3.5 per cent in volume; hence, in ordinary cast iron the graphite will occupy more than one-tenth of the total volume. A cast iron containing elementary carbon, crystallized in the form of graphite plates between crystals of ferrite, is only used to-day when the mechanical and temperature stresses are low. Cast iron for moderate temperatures should have a pearlitic matrix in which the ferrite crystals and the needles of graphite are intermixed (Fig. 98, left). When cast iron comes into contact with hot steam a better quality must be used (Fig. 98, right). Tests and experience have shown that cast iron containing fine plates of primary graphite, uniformly distributed in an exceedingly fine grained pearlitic base, is good for withstanding thermal and mechanical stresses.

Under the influence of high temperatures cast iron begins to transform itself gradually; its volume "grows" sufficiently to be able to cause serious trouble in practice. The influence is all the quicker, the higher the temperature and the more frequently it changes. The ordinary temperatures for live steam at the present day are sufficient to start the transformation. The growing begins with the decomposition of the ferro-carbide into ferrite and elementary carbon in the form of graphite. The carbon already present as graphite plates forms nuclei which grow. Once the disintegration has begun, and the ferro-carbide is decomposing, its completion is only a matter of time. The steam and gases which it containes penetrate into the casting along the graphite needles. Oxygen is the predominant gas, the crystals of metal are oxidized and, finally, the cast iron is completely decayed. Fig. 99 shows very well the progress of the disintegration in the case of a poor quality cast iron which was exposed to a temperature of 617° F. (325° C.) when in service. The matrix of iron is ferrite and it contains a network of graphite crystals. The decay of the structure and the oxydization are proceeding from the left. They follow the graphite needles right into the body of the casting and can be distinguished by their grey colour. It will be noticed that they are already well advanced.

The tendency of cast iron to disintegrate can only be prevented to a small extent by the addition of alloys. Manganese and chromium favour carbonization; chromium also gives cast iron a very fine grain. Silicon, on the other hand, forms

Fig. 99. Decomposition of cast iron. Microphotograph, magnified 100 times
3,3 % C.
2,5 % Si.
0,71% P.
0,13% S.

with iron very stable silico-ferrites, it attracts the iron from the cementite and promotes the segregation of the graphite. Nickel works in this way, but it also increases the hardness of the metallic base. All attempts, therefore, to improve the composition of cast iron by alloys are principally directed to refining the metallic matrix. The improvement of the mass of metal, however, will not have the effect it should as long as the graphite is separated in an undesirable form. For this reason it should be endeavoured at the same time to distribute the graphite in very fine particles, or else to diminish the graphite content by various means. It is only after many years of testing and observation that it can be decided whether it is definitely possible to abolish growing and render cast iron suitable for exposure to steam at high temperatures.

Steam turbine constructors should be thoroughly conversant with these properties of cast iron as it is often used for casings, diaphragms and accessories. Bearing pedestals, bedplates or gear wheels and their casings do not offer the same difficulties because they do not come into contact with live steam or ever get unduly hot. The composition and structure of parts for live steam which are made in cast iron should be accurately specified; it is only in this way that sufficient reliability can be assured after prolonged service. A suitable cast iron will contain about 2.7 to 3.2% of carbon, 1.2 to 1.5% of silicon, 0.8 to 1.2% of manganese, and not more than 0.4% of phosphorus and 0.1% of sulphur. This will enable a bending stress of 21.6 tons per sq. in. (34 kg./mm.^{2}) and 0.39 in. (10 mm.) deflection to be obtained with the standard test bar. The highest steam temperature to which cast iron may be exposed at the present day is 480° F. (250° C.) for a pressure of 285 lb./sq. in. (20 kg./cm.^{2}) gauge. The

temperature may be higher for lower steam pressures but it should never be more than 575° F. (300° C.). All parts, therefore, which are subjected to temperatures greater than the above should be made of cast or forged steel.

The chief concern when casting large exhaust casings of L.P. turbines, with relatively thin walls and ribs, is not so much the quality of the material as the precautions to be taken in the making of the moulds, in the casting and in the cooling down. Cracks due to shrinkage or internal stresses can be very dangerous and, in the case of large castings, rejection is costly and delays considerably the delivery of the turbine.

b. Cast steel

Cast steel contains carbon in hypoeutectoid quantities and always in a combined form. There is no question of any free graphite. Cast steel, therefore, keeps its original structure, it does not grow and does not disintegrate, even at high temperatures. In good quality cast steel it is important that the two undesirable components, phosphorus and sulphur, should be present in as small quantities as possible. This will reduce the risk of them forming localized accumulations. Phosphorus makes a casting coarse grained, brittle and cold-short.

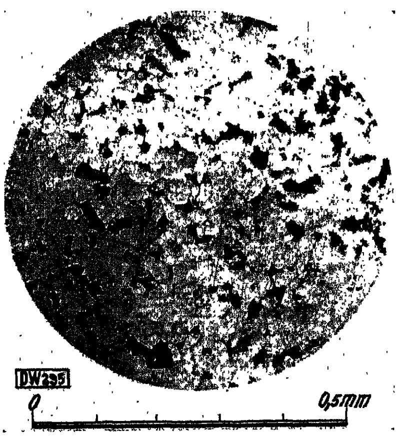


Fig. 100. Microphotograph of cast steel. Magnified 100 times

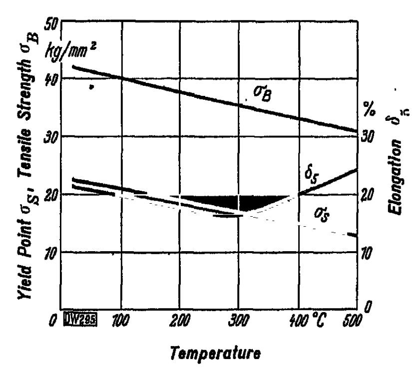


Fig. 101. Tensile tests of unalloyed cast steel

A high sulphur content causes a greater tendency to hot-shortness. At the present time phosphorus and sulphur together should not exceed about 0.12% and there should not be more than 0.07% of either element. There would be no objection, however, to taking somewhat higher values if there were the certainty that the phosphorus and sulphur would be uniformly distributed over the whole casting. The structure of well annealed cast steel (Fig. 100) only shows pearlite (dark) and ferrite (light).

The yield point of cast steel falls by about half between the ordinary temperature and 930° F. (500° C.) as can be seen from Fig. 101. The curves have been especially simplified to render them easily applicable. Cast steel is used chiefly for casings and valve chests of the H.P. part of turbines. These castings are relatively small, and there is usually no difficulty in making the walls thick enough to withstand the stress.

Cast steel shrinks more than cast iron when cooling and for this reason the risk of blow holes and internal stresses is greater. Cast steel is also more flexible, and large deformations may take place in the process of cooling without producing permanent stresses. In addition to this, steel foundries have found means of greatly increasing the quality and the uniformity, also the

methods of testing have improved so much from recent experience that the reliability of steel castings should no longer be doubted.

The heat treatment of cast steel is especially important. The first annealing is at about 1550 to 1650° F. (850 to 900° C.) and recrystalizes the coarse structure; the second one, after the rough machining, relieves the internal stresses. The exact temperature and duration of the operations and the time required for cooling should be determined in every case from results of microscopic examinations. The recrystallized structure will be finer grained the less the A_3 line (ferrite line) is exceeded, the shorter the time during which the solid solution is heated above its temperature of formation, and the quicker the cooling process. If the casting is heated to a too high temperature the grain becomes coarse again, a "superheated structure" similar to the original casting is obtained and can only be eliminated by further heat treatment. The time required will depend on the thickness of the walls and the weight of the casting, an excessively long treatment producing a coarse grain. A certain time is also required for cooling; if it is too short internal stresses will result. The second annealing is essentially for eliminating any internal stresses which may still be present from the casting or which may have been produced when cooling down or during the rough machining. Temperatures up to about 1200° F. (650° C.) are used. It is necessary to remain below the pearlite line so as not to cause any unrequired change in structure; on the other hand, the limit of elasticity must be lowered sufficiently to allow the internal stresses to be relieved.

In connection with plants for higher pressures, reference is often made to cast steels alloyed with nickel or nickel-chromium. Recently a molybdenum alloy has been frequently mentioned. These additions improve the mechanical properties of cast steel, they raise the yield point, the tensile strength and the impact figure. Investigations have shown, however, that in the case of a nickel alloy, for instance, the yield point is only considerably raised when the piece is cold. At 930° F. (500° C.) not more than about 10% improvement is obtained over ordinary cast steel with 32 tons per sq. in. (50 kg./mm.^{2}) strength. In the case of a molybdenum alloy the drop in tensile strength appears to be less. When the temperature is not exceptionally high ordinary cast steel is sufficient, the addition of an alloy meaning a considerably higher price.

c. Mild steel

Unalloyed Siemens-Martin steel is chiefly used in steam turbine plants for making live steam piping. The kinds which have usually been employed up to the present have the ordinary structure of a low carbon steel. The grains are ferrite surrounded by a certain amount of pearlite (Fig. 102, left). This material is on the whole quite satisfactory. *Krupp*'s "Izett" steel, which is less crystalized, and similar steels are better (Fig. 102, right). As a result of special treatment they contain practically no oxygen or nitrogen and have a more uniform structure. These steels differ only slightly in strength, they all have, under similar stresses, about the same resistance as ordinary unalloyed mild steel for live steam piping (Fig. 103).

It has been proposed to use high-grade steel, alloyed with nickel, molybdenum or other substances, for live steam piping. There seems to be no need, however, to employ such materials. The pipes certainly are better and, especially, there is less recrystallization, but the same improvement can be obtained with Izett steel. Needless to say, pipes and materials must always be carefully tested.

Forged Siemens-Martin steel with a tensile strength of 38 to 41 tons per sq. in. (60 to 65 kg./mm.^{2}), a yield point of 22 to 24 tons per sq. in. (35 to 38 kg./mm.^{2}) and 24% elongation on a 5 : 1 test piece, is generally used for rotating parts such as shafts, solid rotors, discs and drums. For parts subjected to high stresses, the steel will contain a small quantity of nickel. Of late, the

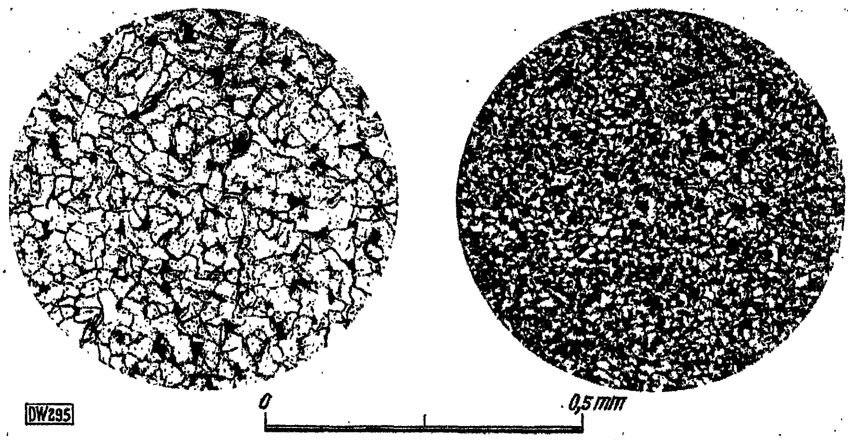


Fig. 102. Microphotographs of materials for H.P. steam piping. Magnified 100 times
Unalloyed S.M. steel "Izett" steel

most highly stressed wheels of the last stages of limit turbines are even made of nickel-chromium steel with 51 to 57 tons per sq. in. (80 to 90 kg./mm.²) tensile strength, about 41 tons per sq. in. (65 kg./mm.²) yield point and 15% elongation on a 5 : 1 test piece (Table I).

As a result of the forging the metal is more tenacious. It should be remembered, however, that the improvement of the mechanical properties takes place chiefly in the direction of the stretching, partly at the expense of the ductility in a perpendicular direction. A forging should be designed and manufactured so that the direction of the main stresses corresponds with the grain of the metal. It is particularly important that the pieces should be forged right through, for if the ductility is insufficient, no subsequent annealing can raise it to its proper value. It may be necessary, for instance, to give a disc a profile differing from the best form, according to calculations, in order to obtain a more uniform quality of metal from the hub to the rim. A disc of uniform strength, having a large amount of metal at its hub, could not be sufficiently forged through and the poorest material would be found at precisely the point of highest stress. A disc of uniform quality of material would have a considerably smaller hub. By forging the discs over a mandril the metal at the hub is also worked from the inside and a better structure is obtained at the joint of the disc and the hub.

The metal at the centre of large forgings is never of such good quality as towards the surface, the slower cooling down of the centre making the metal poreous. For this reason care should be taken that the axis of rotation of a shaft or a rotor always corresponds to the centre line of the forging. The expansion at all points the same distance from the axis of rotation will then be equal, and there will be no distortion when the rotor is heated.

The characteristics of insufficient forging, which may be seen in a large shaft, is a coarse texture formed by a granular sorbite with a fine net of ferrite (Fig. 104, left). If the strength of the metal is satisfactory and the structure of the outer layers up to standard,

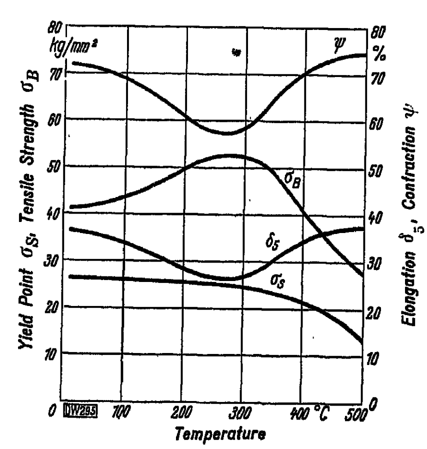


Fig. 103. Tensile tests of unalloyed S.M. steel (H.P. steam piping)

Material		Tensile strength σ_B tons per sq. in. (kg./ mm.²)	Yield point σ_S tons per sq. in. (kg./ mm.²)	Elongation of 5:1 test piece δ_5 %	Applications
Mild steel		38 to 41 (60 to 65)	22 to 24 (35 to 38)	24	Rotors, discs, drums
5% nickel steel		38 to 41 (60 to 65)	27 to 30 (42 to 47)	26 to 22	Blades, bolts for use with superheated steam
3% nickel steel		44 to 48 (70 to 75)	32 to 35 (50 to 55)	18	Highly stressed discs, shrinkage rings
Special S. M. steel		44 to 51 (70 to 80)	29 to 32 (45 to 50)	17	Rings for gear wheel for high stresses
Nickel-chromium steel		51 to 54 (80 to 85)	32 to 33 (50 to 52)	18 to 15 (longitudinal) 12 (tangential)	Pinions for high stresses
Nickel-chromium steel		51 to 57 (80 to 90)	41 (65)	15	Highly stressed discs
Stainless steel	hot rolled	44 to 48 (70 to 75)	32 to 35 (50 to 55)	26 to 22	Turbine blades
	cold rolled	40 to 43 (63 to 68)	27 to 30 (42 to 47)	18	

Table I. Steel for steam turbines. Mechanical properties and applications

the forging may be used. The section shown on the right is from the well-forged outer layer. It would be almost impossible to obtain such a structure right through a large shaft. Forgings are subjected to heat treatment similar to steel castings in order to obtain a finer grain and to relieve machining stresses.

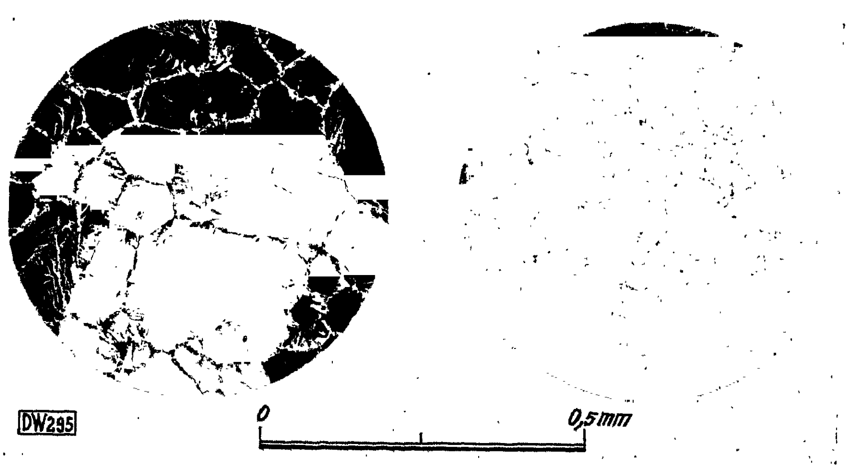


Fig. 104. Micrographs of special S.M. steel (3% Ni). Magnified 100 times

Forgings for withstanding high stress usually undergo further treatment. They are heated slightly above the ferrite line, to between 1470 and 1560° F. (800 to 850° C.) according to the composition of the material, and then quenched in oil. It is necessary to eliminate any stresses due to the tempering, also an

elastic metal, and not a hard one, is required. This is managed by heating the forging again to a relatively high temperature between 930 and 1380°F. (500 and 750° C.). This treatment improves the yield point and the elongation. The chief object of such an expensive treatment, however, is the large increase in the ductility and the impact figure. It may be desired to verify whether the forging has received a proper heat treatment and is free from internal stresses. After the rotor has been completely machined and bladed, it should be suitably heated for a certain time to a temperature of at least 180° F. (100° C.) higher than the maximum temperature when in service. It is then cooled gradually in six to ten hours. During the operation it is made to rotate slowly or, if the necessary plant is available, it is run at overspeed. If no distortions take place during the test, the rotor will probably always remain true during operation.

Rolled mild steel of the ordinary commercial quality is good enough for bolts which are moderately stressed and are not heated above 575° F. (300° C.). For higher temperatures use should be made of Siemens-Martin steel with 32 to 38 tons per sq. in. (50 to 60 kg./mm.²) tensile strength, 22 to 25 tons per sq. in. (35 to 40 kg./mm.²) yield point and 24% elongation at the rupture of a 5 : 1 test piece. Bolts which are still more highly stressed, such as those at the joints of H.P. casings especially, are made of 5% nickel steel with 38 to 41 tons per sq. in. (60 to 65 kg./mm.²) tensile strength, 27 to 30 tons per sq. in. (42 to 47 kg./mm.²) yield point and 26 to 22% elongation. Recently a nickel-chromium steel of special composition has been used with turbines and piping for very high pressures. Ordinary stainless steel has not proved itself suited for this purpose. On the other hand, a steel with 5% nickel, 1.5% chromium and 1% tungsten appears to be quite satisfactory. At the ordinary temperature it has a tensile strength of about 70 to 76 tons per sq. in. (110 to 120 kg./mm.²), a yield point of 60 to 64 tons per sq. in. (95 to 100 kg./mm.²) and 17 to 15% elongation. At 930° F. (500° C.) these figures are still about 53 tons per sq. in. (84 kg./mm.²), 41 tons per sq. in. (65 kg./mm.²) and 18% respectively.

d. Special steels

Special steels, such as high-grade nickel steel and stainless steel, are used in turbines mainly for the blading (37). 5% nickel steel and stainless steel with a small amount of nickel and about 15% chromium are chiefly used. Nickel steels are remarkable for the high value of their yield point relative to their tensile strength and for their toughness. Chromium diminishes to a certain extent the susceptibility to oxydization, and the hard chromium carbides make the steel more resistant against erosion. Of late, steels containing large quantities of nickel have been advertised. They have up to 60% of nickel and about 15% of chromium, as in the cases of the French ATV steel, or the German B7M steel. The chemical composition of these steels approaches that of Monel Metal. This latter material, when pure, is well suited for

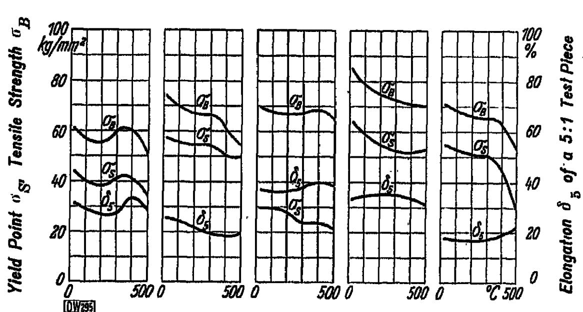


Fig. 105. Tensile tests with different blade materials

5% nickel steel	V5M steel	ATV steel	B7M steel	Monel Metal
%	%	%	%	%
5.0 Ni.	0.7 Ni.	34 Ni.	60 Ni.	67 Ni.
0.4 Mn.	14 Cr.	11 Cr.	14 Cr.	28 Cu.
0.15 C.	0.5 Mn.	1.0 Mn.	6.8 Mo.	5 Fe.+Mn.
0.2 Si.	0.16 C.	0.42 C.	1.4 Mn.	Traces Si.+C.
0.02 S.	0.6 Si.	0.15 Si.	0.03 C.	
0.02 P.	0.02 S.	0.03. S.	0 95 Si.	
	0.008 P.	0.03 P.	0.02 S.	
			0.01 P.	

(37) See *H. Krüger:* "Dampfturbinenschaufeln" (Berlin : J. Springer 1930).

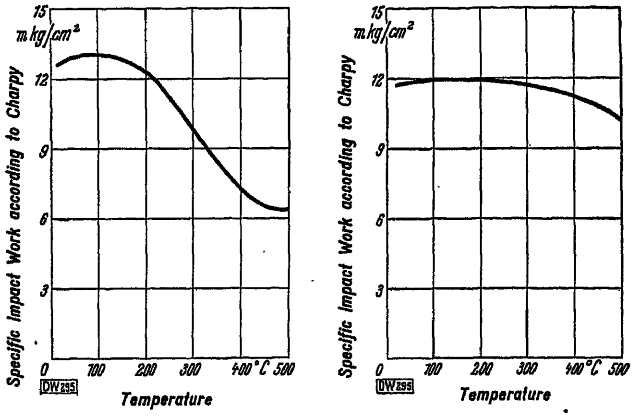


Fig. 106. Notched bar impact tests of 5 % nickel steel (left) and V5M stainless steel (right)

making blading; it is unfortunately not always reliable, and is very expensive.

The tensile strength and yield point of 5% nickel steel fall relatively rapidly as the temperature increases, they fall less precipitately in the case of stainless V5M steel (Fig. 105). The curve for 5% nickel steel shows a tendency to blue-brittleness, which is, however, very small. This dip is quite absent in the case of V5M steel. Notched bar impact tests with these two steels for temperatures up to 930° F. (500° C.) show a falling off of the impact figure after about 400° F. (200° C.) for nickel steel (Fig. 106), whilst for V5M steel there is hardly any decrease up to 930° F. (500° C.). A good impact figure means less liability to excessive overstresses and it also gives very important information relative to the "health" of the material.

Tests have been made with these two steels to determine whether a deformation in the cold state has any influence on the toughness at higher temperatures. Test pieces were elongated 8% when cold, impact tests were made and the figure was established. As far as the brittleness or tenacity can be ascertained from notched bar tests, it would appear from the curves in Fig. 107 that there is no appreciable blue-brittleness for either 5% nickel steel or stainless V5M steel. A comparison with the tests made in the usual way shows a diminution of the tenacity, which is doubtlessly due to the cold deformation. The temperature at which the tests were made has no particular influence. There is, especially, no marked change in the shape of the curves for specimens which have been stretched when cold.

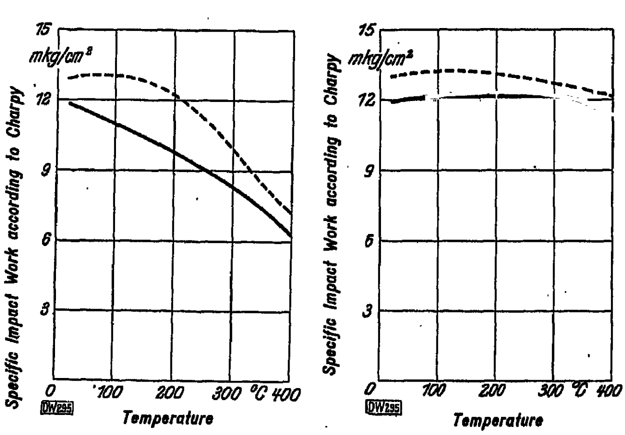


Fig. 107. Notched bar impact tests of 5 % nickel steel and V5M stainless steel, previously elongated 8% at 68° F. (20° C.)

------- not deformed ——— cold deformed

5% nickel steel		V5M steel
39.4 (62.0)	Tensile strength, tons/sq. in. (kg./mm.²)	45.0 (70.9)
30.0 (47.2)	Yield point, tons/sq. in. (kg./mm.²)	36.1 (56.9)
31.0	Elongation of a 5 : 1 test piece, %	25.0
71.0	Contraction, %	68.0

If the physical properties of the two steels are compared it will be found that V5M steel il only superior to 5% nickel steel from 575° F. (300° C.) onwards. At low temperatures the higher yield point and tensile strength of V5M steel is compensated by the lower values of the elongation and the impact figure. The strength or elongation of these two steels is not equal to those of either ATV or B7M steel which, however, have only been used

so far in a few experimental cases. B7M steel has also a higher yield point. Unfortunately, these good physical properties can only be obtained for a high price. B7M steel, for instance, costs about five times as much as V5M steel, whilst Monel Metal costs three times as much.

The value of a steel cannot, however, be definitely settled by its physical properties alone. These have to be considered with the micro-structure especially, to which, in turn, must be compared the other characteristics. 5% nickel steel (Fig. 108) has a grained pearlite structure of ferro-carbide imbedded in ferrite. It is heat-resisting. A prolonged heating at 750° F. (400° C.) causes no change in structure. V5M steel has a very similar appearance (Fig. 109), although a certain direction of stretching and some small enclosures of slag and double carbide are visible. It also is not affected in structure after being heated to 750° F. (400° C.). ATV high-grade carbon steel contains numerous globular carbides and has a tendency to form twin grains, but otherwise it has a very uniform structure (Fig. 110). The specimen had been heated up to 1560° F. (850° C.). The structure has not changed from its original

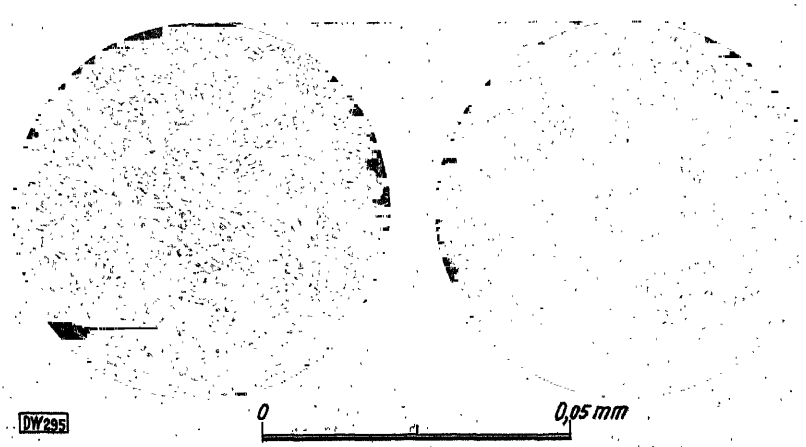


Fig. 108. Microphotographs of 5% nickel steel. Magnified 700 times
As delivered — After being heated to 750° F. (400° C.)

state; this steel should, therefore, be able to withstand even brazing, as might be employed for fixing the coverband to the blades or the blades to the rotor. B7M high-grade nickel steel has also, on the whole, a regular single-phase polyhedral structure as is usually seen in high alloy steels (Fig. 111). The granulation is medium sized. The specimen shown was heated up to 1560° F. (850° C.) during one hour; it was then left to cool in free air and, finally, warmed up to 570° F. (300° C.). The structure is hardly altered. A few small enclosures of slag may also be seen in this specimen. Such enclosures are usually to be found in Monel Metal in much larger quantities as shown on the left of Fig. 112. The specimen on the right is from a sound piece.

A chromium-vanadium steel has sometimes been used in America for moving blades. According to the opinion of several steel producers its properties would not be superior to those of nickel-chromium steel. The reason for it being preferred to nickel steel is most likely the low price of vanadium in America.

Experiments for obtaining the necessary resistance against corrosion and erosion by means of metallic coatings (aluminium, nickel, chromium, zinc and brass) have not so far given any encouraging results. In spite of this they are still being continued in the laboratories of certain producers and users.

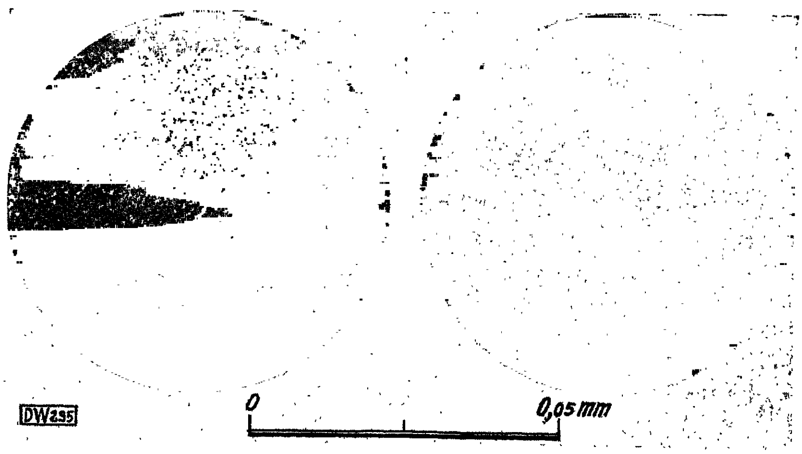


Fig. 109. Microphotographs of V5M stainless steel. Magnified 700 times

As delivered

After being heated to 750° F. (400° C.)

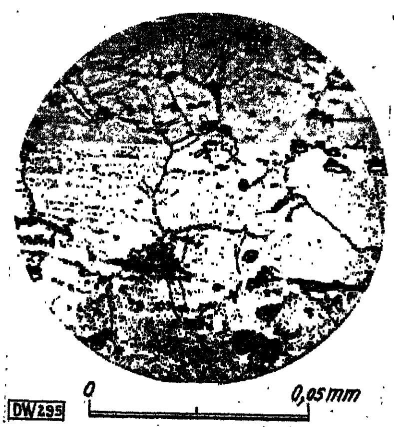


Fig. 110. Microphotograph of ATV stainless steel. Magnified 700 times

After being heated to 1560° F. (850°C.)

Fig. 111. Microphotograph of B7M stainless steel. Magnified 100 times

Annealed at 1560° F. (850° C.), cooled in free air

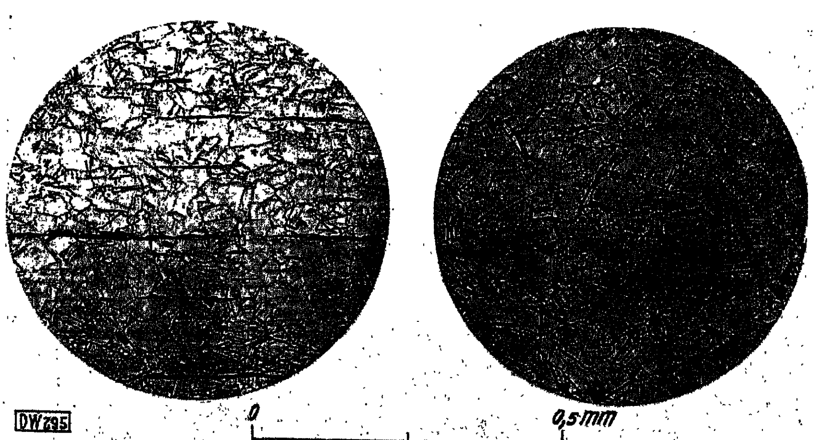


Fig. 112. Microphotographs of Monel Metal. Magnified 100 times

e. Non-ferrous metals

Non-ferrous metals are widely used for glands, for bearing linings and for condenser tubes.

A brass alloy containing 5% nickel, 42% zinc and 53% copper has proved satisfactory for labyrinth packing fins up to 930° F. (500° C.). It should contain neither lead nor tin which might melt out at high temperatures. Monel Metal is difficult to machine owing to its toughness, and it might injure the shaft if it came into contact with it. For these reasons it is not suited for this purpose. Some American firms make the moving and stationary fins of carefully annealed steel. The gland boxes are made of cast iron, cast steel, or they may be of the same material as the fins and rings in order to have the same expansion. It would not be correct to use cast iron for glands at high temperatures.

A white metal with a large content of tin (about 80%) is satisfactory for bearings. It is usual to employ the same metal for journal and for thrust bearings. It is often said that the lead oxidizes when it is in large quantities in white metal. This is not the case, however, it is the two hardening components, calcium and sodium, which oxidize. This explains why two kinds of white metal, Calcium metal and Lurgi metal, which are very satisfactory for many types of machines, are not suited for use with forced lubrication. Thermit, which has also a large content of lead, seems to be more satisfactory.

Condenser tubes are mostly manufactured out of brass containing 70% copper, 29% zinc and 1% tin and having a tensile strength of 22 to 29 tons per sq. in. (35 to 45 kg./mm.^{2}). It is important that the tubes should be free from internal stresses; they should, therefore, be carefully tempered. They must be capable of being flattened without cracking and when they are sawn open they must retain their shape, neither opening out nor rolling up. After many years experience and with most careful testing and treatment, failures of condenser tubes still occur occasionally. This seems to suggest that the root of the trouble is not wholly in the material of the tubes. The British Admiralty believe that they have found a reliable material in an alloy of 80% copper and 20% nickel. It is much harder than the usual brass alloy, it is also more expensive. Its indisputable superiority has not yet been proved.

2. Testing of the materials

It is only by means of careful investigations of the materials of all important parts that engineers may be certain of the presence of the qualities on which the calculations have been based. The results of tests may depend on the way they are taken, on the shape of the test piece and the cutting out from the main piece, or on other factors. A definite agreement must be arrived at between producer and user as to the test methods, the shape and position of the specimens and the manner of working out the results.

The question of the relation between the duration of the test and the values of the strength obtained is still very far from settled and no final and general solution can be given. All figures which are stated here will be referred to the case where the rupture takes place in about three minutes. The designer should, naturally, employ the usual safety factors, always taking those values which correspond to the particular operating temperature.

a. Methods of testing

The methods of testing the different parts in a steam turbine must vary according to the nature of the stresses. Physical, chemical, microscopic and mechanical tests are the most important.

It is by physical tests that such questions as the heat expansion of the materials for packing glands are investigated. The fixed and the rotating parts

of glands should have, as near as possible, the same expansion, otherwise the small radial and axial play and the high temperature of the throttled steam might easily cause the two surfaces to rub and the usual consequences would follow. The value of the chemical research of the composition of materials is most apparent in the case of the analysis of cast iron. The amount of manganese and silicon gives conclusive information on the hardness and on the proportion between the graphite and carbide contents. An excess of phosphorus or sulphur can also be detected by analysis. Microscopic investigations are closely allied to other investigations of the composition. They show irregularities in the structure due to blow holes and enclosures of slag; they allow the structure of the crystals to be examined and conclusions may be formed on such matters as the previous heat and mechanical treatments. The value of the tests which have just been mentioned is usually less in the individual results than in the information obtained concerning the strength of the materials and in the explanations or verifications of the results of mechanical tests. In particular, the values of the strength obtained directly by tests might be very dissimilar and a strict application would mean the refusal of the material. This should not be done, however, before the unsatisfactory results have been compared with those of tests by the other means.

Mechanical tests usually consist in the determination of the tensile strength, the yield point, the elongation, the ductility and the impact figure when cold or when hot, according to the temperature to which the material will be exposed when in service. If no special test pieces have been provided the hardness may be tested on the finished piece and the approximate strength of the material may be determined by means of empirical coefficients. The notched bar test is particularly useful for steels which have had a mechanical or heat treatment. It gives information concerning the nature and extent of the treatment and from its results the purity of the material may also be determined; enclosures of slag and blow holes, for instance, have the same effect as cavities and reduce the toughness.

All these tests, however, do not definitely show whether the material will retain the requisite properties after long service. The "life" of the material must be considered as a further property. It has become essential in the case of highly stressed parts, with a constant or a varying load, that it should be known by the designer. It is well known that steel under stress, even below the yield point, turns into a plastic state, and it will most certainly give when under a high temperature; it "grows" or "flows". This fact is the cause of the discrepancies of the results of different tests on the rupture of the same material, the values differ according to the speed with which they have been obtained. Systematic investigations have shown that there is a maximum stress which the material is capable of resisting for any length of time, after a certain permanent deformation has taken place. Above this "creep limit" the deformation will proceed gradually, until rupture occurs; below the limit recrystallization causes the creeping slowly to cease. In the case of H.P. casings, these properties are particularly important as the cast steel may be exposed to very high pressure and temperature stresses. The yield point and creep limit are very far apart between 750 and 930° F.

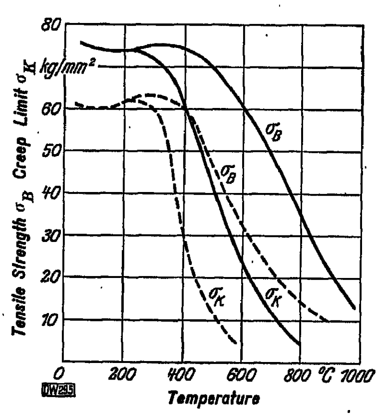
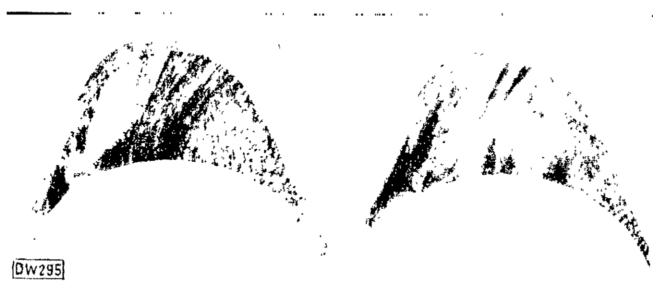


Fig. 113. Tensile strength σ_B and creep limit σ_K of nickel-chromium steel (———) and S.M. steel with 0.35% C. (------) as functions of the temperature

(400 and 500° C.) for unalloyed cast steel (Fig. 113) (38). If it is difficult to make the walls thick enough to reduce the stresses, the creep limit may be raised by using alloyed steels. For temperatures such as may be used at the present day for turbines, alloyed steels are certainly less liable to creep than ordinary steels.

Fracture from an overstress



Fatigue fracture due to vibrations

Fig. 114. Appearance of fractures of mechanically destroyed blades

Alternating stresses offer a very different problem to that of constant loads. The stresses vary persistently between two limits (39). The case occurs, for instance, in all moving blades, the steam jet strikes the blades periodically, the discontinuity being caused by the thickness of the nozzle walls or guide blades. In this way the stress will always be changing. The blades have their own period of vibration and, as they receive impulses at regular intervals, there is a risk of resonance. Stages with partial admission are particularly exposed to this danger. The unceasing changing of the stress will finally fatigue the material and rupture may occur, even if the

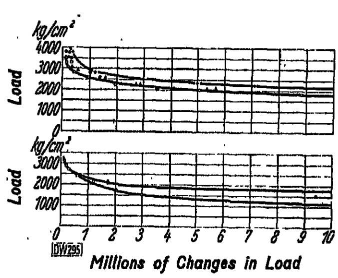


Fig. 115. Results of endurance tests of S.M. steel (top) and 25% nickel steel (bottom) under 3000 changes of load per minute and with the load varying between the maximum tension and compression

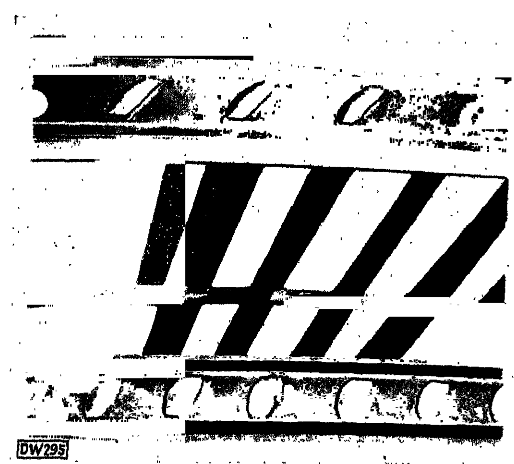


Fig. 116. L.P. blades of stainless steel eroded by wet steam

(38) According to *A. L. Mellanby* and *W. Kerr:* "The use and economy of high-pressure steam plants", Engineering 123 (1927) p. 118 Fig. 2.

(39) Refer to the Fachheft Dauerbruch of the Zeitschrift für Metallkunde 1928. An extract will be found in the Z. VdI. 72 (1928) p. 537 (*P. Melchior).*

stress remains well below the tensile strength which has been determined by ordinary methods. Such a vibration or endurance fracture is characterized by its generic shelly fracture (Fig. 114) and can always be distinguished from an ordinary rupture. It is not sufficient, therefore, to judge the quality of a blade material from the results of ordinary tensile and notched bar tests only, the endurance strength should be the basis for fixing the permissible stress. It has been found, in practice, that if steel is going to break under the influence of repeated stresses, it will do so before five to ten million changes in load. It will not break once this number of impulses has been surpassed. The larger the stress, the smaller will be the number of changes to produce rupture (Fig. 115) (40). The highest stress which may be repeated ten million times without causing rupture is called the fatigue limit. The rapidity with which the load is changed is almost of no consequence. If the changes in section are very gradual and well rounded off, the risk of the effects of notches is reduced and the probability of a fatigue rupture is diminished. Such trifling causes as minute irregularities in the material, small blow holes which may act as notches, marks of corrosion on a smooth surface or even scratches from machining may produce local increases in stress which the material may not be able to withstand over a certain time. The influence of heat treatment on the endurance strength is similar to that on the tensile strength and on the yield point.

The endurance strength is exceedingly important in the construction of turbines, yet the tests by which it has so far been determined are of little use for proving the suitability of a material for continuous service. It would be of great value if simple methods could be devised which would give sufficiently reliable information on the behaviour of materials under continuous stresses. A rough idea on the endurance strength of steels may be obtained from the formula of *Stribeck* (41) according to which it is equal to 0.285 of the sum of the tensile strength and the yield point.

There are still other types of stresses concerning which no reliable data can be obtained by any of the above methods. In the case of steam turbines there is the erosion of the moving blades, which is usually connected with corrosion. There is also the attacking of condenser tubes which has not yet been fully explained. Even if the best materials are used erosion marks may appear on moving blades of L.P. stages, often after very short service (Fig. 116). The cause is undoubtedly due mainly to the action of the small drops of water contained in the steam. It is a remarkable fact that similar effects have also been observed occasionally in the H.P. region. It is not yet possible to explain with certainty the cause of such occurrences. In this particular case particles of rust and water were probably carried along by the live steam. They would have a much higher velocity than in the L.P. part on account of the denser steam and would, therefore, hit the blades on the inside and not on the back. None of the methods of test which have been mentioned gives a reliable value for the resistance of a material against erosion. It is only by experience or by special tests on blades in a flowing medium that information may be obtained.

Examinations which do not extend over the entire piece naturally only give data for isolated points from which may be concluded the probability or improbability of the proper behaviour of the part. They do not eliminate the necessity for the engineers and workmen to be always on the watch and to report immediately any irregular or unusual condition, such as hard spots, pin holes or cracks. These may then be traced.

---

(40) According to *O. Lasche - W. Kieser:* "Materials and design in turbo-generator plant", Figs. 54 and 55 (London: Oliver and Boyd 1927).

(41) Z. VdI. 67 (1923) p. 631.

Hydraulic pressure is very commonly used for testing a whole part. All casings, fittings and condensers, for instance, are examined in this way in order to discover any porous places in the walls or any leakages at the joint. The usual conditions for the test are one and a half times the working pressure and atmospheric temperature. Hydraulic tests cannot be used for parts exposed to high temperatures. A casing may be designed for a factor of safety of about 2.5 referred to the yield point at 930° F. (500° C.). If it is subjected to its ordinary load when cold it will be only very moderately stressed, the material being much more resistant. In this example, therefore, the usual hydraulic test would not be sufficient for judging the tightness as it does not in the least represent the actual conditions and gives no evidence as to the behaviour of the piece at high temperatures. If it is wished to reproduce approximately the actual working conditions a pressure several times greater would have to be used for the test. As far as the casing is concerned this could probably be done but the bolts at the joint, not being at such a high temperature, would be designed for a factor of safety referred to their normal temperature and would be overstressed during test. In such cases the only solution is to abandon the cheap, convenient and undangerous method of hydraulic testing and to use steam at the proper temperature.

b. Test pieces

It is not possible for the material of a large part to have the same properties at every point and for this reason it should be carefully considered how and where test pieces are to be taken. They should be taken, generally and in the first place, where the stress is highest according to calculation; secondly, where it has been found by experience that the material is weakest with the particular method of manufacture.

In every important casting provision should be made for at least one test piece which is to be cut out before the part is machined and examined in a laboratory. It is a bad method to cast the test pieces separately. This used to be employed frequently and it is sometimes seen even now. The different rates of cooling might produce very dissimilar strengths and structures of the material. It would be just as wrong, however, to arrange the test pieces to project at any point. Lugs, such as shown on the left and at the bottom of the H.P. turbine casing in Fig. 117, are rarely free from irregularities. The molten iron will form eddies in the flow and will contain, for instance, slag or sand. The test pieces will, therefore, give different values from those of the actual piece. The only place suitable for taking specimens is from a very gradual projection, having the same thickness as the walls of the casting of which it forms an integral part. For turbine casings the pieces might be arranged as shown on the right of Fig. 117, being always near the horizontal joint.

Fig. 117. Wrong and right arrangement of test pieces on a turbine casing of cast steel

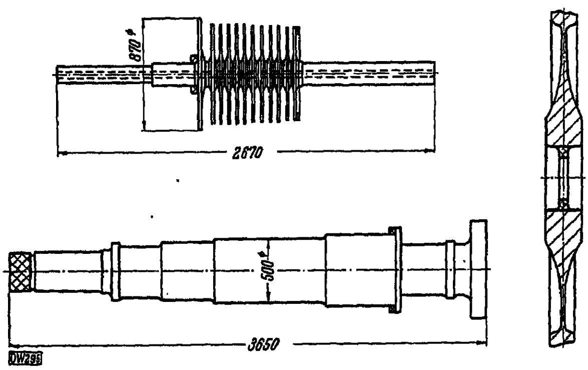


Fig. 118. Arrangement of test pieces on rotors, shafts and discs (dimensions in mm.)

The same care is required when taking test pieces from forgings. As has already been mentioned, the properties of a forging do not depend only on the composition, purity and heat treatment of the material, but they are largely determined by the method and thoroughness of the process of forging. These will determine the arrangement of the grain and the directions of greatest strength and, according to their inclination, the test pieces may give different values.

The proper arrangement is shown in Fig. 118 for an ordinary shaft, a solid rotor and a plain disc. In the case of the ordinary shaft the thorough and even process of forging will have produced a very pronounced direction of the fibres (longitudinal direction). As the diameter of the shaft does not increase by any large steps and irregularities may easily be avoided, and as the stress is usually only small, it will be sufficient to take tests in a longitudinal direction on a sample from the end of the shaft. In the case of larger forgings a tangential test will be taken at the largest diameter (perpendicular direction). The reason for taking tests across the grain, although the shaft is only moderately stressed in that direction, is not to establish differences in the values of the strength which nearly always occur in such large forgings; it is chiefly to get an idea of the structure of the material. In the case of longitudinal tests taken on pieces from the ends of the shaft which have been well forged through, such faults as prevailing cavities and inclusions have been pressed out in the direction of the stress. When taking transverse tests, however, the impurities form lines across the section of fracture and have a great influence. In this way, tests across the grain give better information on the structure of the material.

Two test pieces must always be provided in the case of a solid rotor with serveral large steps in diameter. Apart from the stresses in the shaft ends in a longitudinal and tangential direction, high stresses will occur in radial and tangential directions in the thick part or the discs. Where the parts of small and large diameters meet, a concentration of stresses takes place; in other words, the distribution of the stresses is less favourable than when the change in diameter is gradual, which, as a rule, is not practicable. Experience has also shown that the forging of sharp changes in diameter produces relatively poor values of strength. It would not be correct in this case to take longitudinal tests on pieces from the end of the rotor and it would not be justified only to take transverse tests at the periphery of the discs. According to the principles already stated, tests should be taken at the point where the rotor widens out. If the material at this place and at the periphery is satisfactory the forging may be machined. Shafts which have been bored through may have the walls of the hole examined with a periscope and if the material is found to be good an additional check is obtained. Other methods of examining the interior of forgings have, unfortunately, not yet been developed beyond the experimental stage. For this reason shrinkage holes and large cavities are still feared as they cannot be avoided with certainty even by the most careful manufacture.

In a disc the hub is known to be the most highly stressed part and it is also the most difficult part to manufacture. It has, therefore, been found sufficient to take only a tangential test inside the bore.

In the case of a double-flow L.P. drum the test pieces should be taken as shown in Fig. 119. The arrangement is in agreement with the above principles. It might be considered advisable to take tests inside the bore at both ends of the hollow drum. The two tests on the outside should, however, be sufficient.

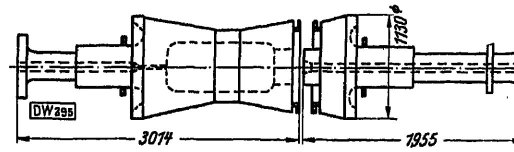


Fig. 119.
Arrangement of test pieces (▨)
on a double-flow,
drum-type rotor in two parts
(dimensions in mm.)

The strength of blade materials is ascertained from test pieces taken from the bars as delivered. It is often well to check the properties of the finished blade. Tensile or impact tests always cause the destruction of the specimen; therefore, only hardness tests may be taken. The value of the hardness is only to secure an idea on the uniformity of the consignment. The calculation of the tensile strength from the hardness is always somewhat unreliable. No conclusion may be arrived at concerning the yield point.

3. Stresses

The stresses in all parts of a construction should be kept as small as possible in order to satisfy the condition of absolute reliability. Above all, the safety factor relative to the yield point should not be less than 2.5. The stresses should be kept especially low in parts where vibrations and the resulting fatigue are feared.

Moving blades are subjected to tension and bending under the centrifugal force and the steam pressure and, if their dimensions are not sufficient, the reliability of operation may suffer. Blade fractures may begin in the cross section of the root and neck or from the bearing surfaces of the root. There are many ways of avoiding the danger. The tension and the bending moments of the centrifugal forces may be reduced by thinning or tapering the blades towards their tips and by mounting them so that the axis is radial. The bearing surfaces may be augmented by using blades with integral packing pieces, by strengthening the root on one or both sides or by using double grooves (Fig. 41). Serrated grooves (Fig. 42 *g* and *k*) are very favourable in this respect, especially for limit turbines. There are many bearing surfaces and the axial distances between them are relatively large. When manufactured accurately the bearing pressure can be considerably reduced and the blade will be less liable to axial vibrations. The changes in section should be rounded off as much as possible as particularly dangerous local stresses may occur at sharp edges or angles, reaching sometimes two or three times the normal value.

The stress which may be allowed will vary according to the material. At the maximum speed of rotation steel blades should not be stressed beyond 12.7 tons per sq. in. (2000 kg./cm.^{2}) at the neck and the bearing pressure should not exceed 15.9 to 17.5 tons per sq. in. (2500 to 2750 kg./cm.^{2}).

The following table shows, as well as is possible from present experience, the best blade material to choose for various stresses and temperatures:

General use for H.P. stages up to 525° F. (275° C.) . .	Low percentage nickel steel
For temperatures above 525° F. (275° C.) and stresses greater than 1.9 tons per sq. in. (300 kg./cm.^{2}) . . .	Stainless steel
For L.P. stages below 400° F. (200° C.) and 3.2 tons per sq. in. (500 kg./cm.^{2})	Brass
For L.P. stages below 400° F. (200° C.) and 6.4 tons per sq. in. (1000 kg./cm.^{2})	Nickel-brass
Over 6.4 tons per sq. in. (1000 kg./cm.^{2}) and generally over 400° F. (200° C.)	Low percentage nickel steel, stainless steel or Monel Metal.

The high peripheral velocities, especially in the L.P. part, require the stresses to be accurately calculated in the coverbands. A good way of considering the question is to assume the coverband to be an elastic strip held in place by the rivet tenons. For this unfavourable calculation the stresses should not exceed 16 tons per sq. in. (2500 kg./cm.^{2}) with steel or Monel Metal, or 6.4 tons per sq. in. (1000 kg./cm.^{2}) with brass. It may not be possible to obtain these values over the locking piece; the coverband should then be lightened by drilling out a hole about the middle between two blades the hole extending over a third of the coverband width and a half the blade pitch; if the stress is still too high, the coverband may be entirely suppressed over the locking piece. This solution should not be adopted if it can be avoided as not only will there be a large gap in the blading, but the two last blades not being so well secured, there is greater danger of axial vibrations. The risk is reduced if the groups of blades are joined by lacing wires. When the clearance is greater than 0.12 in. (3 mm.) the coverband may be made of the blade material. For small clearance brass or Monel Metal may be used as there is the possibility of rubbing occuring. Highly stressed coverbands should be of low percentage nickel steel or stainless steel.

Special care must also be paid to the rivet tenons at the tips of the blades. When the peripheral velocity is great they may be overloaded by the coverband and their section should be increased. This may be done either by reinforcing the blade by a small projection, tapering off towards the foot, or by widening the whole profile at the tip. When the second method is employed the blades would certainly be more heavily loaded at their base.

Discs for high stresses are profiled, those for low stresses are usually of constant thickness. Dangerous stresses may occur in the rim, at the section of minimum thickness, at the bore of the hub and around the pressure equalizing holes. It is, therefore, of first importance that the width of the rim should be sufficient even if this increases the stresses in the hub. The chief stresses in the rim are due to the centrifugal forces of the blades which tend to make it burst open. If the rim were to expand, the blades might become loose in the grooves and there would be a greater tendency for them to vibrate. Another advantage of a small widening of the rim is that the possibility of disc vibrations is greatly reduced. It is also necessary to avoid too great a difference in thickness between the rim and the tip of the disc. There would otherwise be the danger of the wheel bursting, the thin disc not being able to resist the expansion of the heavy rim. If there are any equalizing holes, their edges should be rounded off as it has been found that most failures from fatigue start from holes with sharp edges. For reasons of reliability the stress should not be higher than 9.5 tons per sq. in. (1500 kg./cm.^{2}) for a material with 38 tons per sq. in. (60 kg./mm.^{2}) tensile strength or more than 12.7 tons per sq. in. (2000 kg./cm.^{2}) for 44 tons per sq. in. (70 kg./mm.^{2}) tensile strength, or more than about 15.2 tons per sq. in. (2400 kg./cm.^{2}) for a very good quality material with 51 tons per sq. in. (80 kg./mm.^{2}) tensile strength. If the stresses at the equalizing holes are calculated according to *Stodola*'s method (42), their values may be about 25% higher than is usually allowed as they are only localized.

The danger of the presence of shrinkage cavities is particularly marked in hollow or solid drums. There may also be internal stresses inside the forging produced during manufacture and no information can be obtained as to their values. They add themselves to the normal stresses and may greatly reduce the safety factor below the value according to ordinary calculations. Shrinkage holes and internal stresses are especially to be feared at the point where the drum is connected to the shaft ends as the ingot has to be extensively forged at these places. It is, therefore, necessary, on the one hand, to reduce the risk of fracture

(42) 6th (German) Edition p. 320; English Edition p. 382.

by avoiding forgings of great thickness and complicated shape which are difficult to produce and, on the other hand, to prove the utility of the forgings by means of a prolonged run of the unbladed drum at about 30% overspeed. The stresses are usually very high around the dowel holes at the ends of the drums and they may only be considered safe if there is sufficient material to transmit them. The stress round the bore of a disc and at the shrinkage joints should have at least the ordinary 2.5 safety factor.

Reliability is also of primary importance for stationary parts of turbines, especially for diaphragms and casings.

Diaphragms for pressure differences up to 70 lb./sq. in. (5 kg./cm.^{2}) can be in cast iron, for higher pressures they are made of cast or Siemens-Martin steel. As the present tendency is to use a large number of stages, diaphragms can usually be made as thin flat plates without ribs. The stresses and deflections may be calculated by comparing them to those of flat circular plates by means of experimental coefficients. By this method the essential factor will usually be the deflection when mild steel is used, or the stress when the diaphragms are cast. Stresses in cast iron diaphragms of 3.2 up to 3.8 tons per sq. in. (500 to 600 kg./cm.^{2}) may be allowed.

Certain rules must be observed when designing casings in order to prevent overstresses. It is most important that the casing should be as cylindrical as possible and that there should be no abrupt changes in section. Large accumulations of metal should be avoided on account of heat expansion stresses. The cap nuts should be placed as near as possible to the cylinder and if the flanges at the horizontal joint are thick and narrow, instead of thin and wide, they will be less stressed. Regarding the thickness of the walls, it should be remembered, as has already been said, that the yield point decreases at high temperatures. Hydraulic tests are indispensable as the calculation of a casing by means of formulae, which are often very complicated, is only approximate, especially if there are many openings.

A new method of American origin has recently been introduced for obtaining a tight horizontal joint. The bolts are warmed by an electric heater, the nut is screwed on and tightened a definite amount and the bolts are allowed to cool. By this means sufficient tension is obtained in the bolt for pressing the flanges together. Much higher stresses may be obtained than with ordinary spanners; care should be taken, however, that the stress in the bolts is not more than about one-third of the yield tension. The angle through which the nuts are turned after they have nipped the flange should be accurately determined.

As a result of the ever increasing steam temperatures, greater attention than formerly has to be given to the piping. It should not only be calculated to resist the steam pressure but, if the ends are fixed, the stresses due to heat expansion should be determined. Special care must be given to steam bends which are rendered stiff by thick walls. In this case the expansion will not be compensated by the cross section taking an elliptical shape as happens to a large extent in thin walled pipes. The ability of the bends to take up the expansion may be considerable, it may not be sufficient, however, to prevent the stresses rising above the permissible limit for the hot pipes. Wide loops will then have to be provided for the expansion.

The general conditions determining the reliability have now been given and a more special source of danger may be discussed: it is vibration. Its importance for modern steam turbines is very great and may largely influence the design.

There is always the possibility of vibrations when dealing with a system capable of oscillating in the presence of periodic impulses. If the frequency of the exciting force coincides with that of vibration of the system there will be resonance. It is only when this occurs that large deflections will arise, producing sufficiently high stresses to cause failure by fatigue. In order to obtain complete

reliability it is necessary to take precautions to prevent resonance in any system capable of vibrating.

The most important systems of this kind are the moving blades which are hit by the steam jet. The steam flow is not fully uniform around the circumference, it is interrupted by the thickness of nozzle walls. The blades are all subjected to periodic forces of greatly varying intensity and vibrations may occur. As has already been said, the danger is particularly great in the case of stages with partial admission. The peripheral forces in these stages are usually very high and the static bending stress produced by the steam should not exceed 4.8 tons per sq. in. (750 kg./cm.^{2}) in the case of maximum blade load. It is, of course, necessary to consider the possibility of vibrations being produced by the steam jet in stages with full admission also.

There are other causes which may produce blade vibrations. Periodic forces occur at the horizontal joint, at points where exhaust steam from auxiliary sets is admitted, at bleeder branches or, in general, at any place where the flow is disturbed. Vibrations may occur in a tangential or an axial direction. They may be largely avoided by serveral means. By a suitable design it is possible to alter the natural frequency of the blade or the frequency of the impulses; or to avoid the impulses as far as is feasible. A third method, which has not yet proved satisfactory for blades, is to reduce the amplitude by damping, even when resonance occurs.

The use of lacing wires or of connecting the ends of the coverband is an application of the first method. This means prevents torsional vibrations of groups of blades around a radial axis. It secures the blades at the end of the segments, which are in a particularly dangerous position. It can be recommended for blades from 2 in. (50 mm.) up to the length from which lacing wires are used, when it becomes unnecessary. Other applications of the first method are to strengthen the blade, especially at the foot, to reduce the thickness of the blade edges or to taper the section towards the tip. The effects may be calculated according to methods of *Stodola* (43), *Hort* (44) or *Schwerin* (45).

Applications of the second method are the co-ordination of the number of nozzles to that of the groups of moving blades, or of the number of fixed to the number of moving blades. The walls of the nozzles may be tapered at the exit to obtain a continuous jet and a uniform field of force. It is usually advisable to raise the lowest natural frequency of blades, by one or several of the previous methods, to over 3.5 times the number of revolutions per second. Care must also be taken to avoid resonance between harmonics of blades and the fundamental impulse of the nozzles or its harmonics. Such resonance may also be dangerous.

Laboratories for investigating vibrations allow the results of calculations to be verified and are a great help for obtaining a good design. They are to be found in most of the leading works for manufacturing turbines.

Apart from moving blades, discs are very liable to axial vibrations and the safety of operation may be greatly endangered, a failure being generally very destructive (46). In disc vibrations there are usually some nodal diameters which remain stationary. In order to reduce the possibility of resonance care must be taken that the natural frequency of the disc divided by the number of nodal diameters is not equal to the speed of rotation. Experience has shown

---

(43) 6th (German) Edition p. 946; English Edition p. 1143.

(44) *W. Hort:* "Die Differentialgleichungen des Ingenieurs", 2nd Edition pp. 376/83 (Berlin: J. Springer 1925).

(45) Zeitschr. f. techn. Physik 8 (1927) p. 264.

(46) See *W. Campbell:* "Protection of steam turbine disk wheels from axial vibration", Amer. Soc. Mech. Eng. 1924.

that it is best to use only discs which are so stiff that their critical number of vibration, as defined above, is far in excess of the speed of rotation. Discs with frequencies below the turbine speed should not be used. This method of eliminating disc vibrations has frequently been applied in the H.P. part of recent turbines. All the stages, including the first one, which is generally used for regulation, are of relatively small diameter and sufficient stiffening may be obtained by a slight increase of the minimum thickness of the wheel near the rim or of the hub. It is more difficult, however, to protect the L.P. wheels from vibration, they are mostly of large diameter and have very long blades.

Shafts are also liable to vibrate and the running may be affected by transverse or sometimes torsional vibrations.

Transverse vibrations are mainly due to imperfect balance, the unsymmetrical centrifugal forces creating vibrations. Therefore, the best way by which smooth running may be obtained, even if the running speed is near the natural frequency, is accurate balancing. The running of the turbine may also be affected by an unsuitable form of bearing surface. According to *Stodola* (47), the cause would be the elasticity of the film of oil. Care must be taken, therefore, that the natural frequency of each separate length of shaft for the correct temperature should not be in phase with the speed of rotation. Neither must this happen to any of the combined frequencies of shafts with several spans. The frequencies of a continuous shaft may be calculated according to the method of *Stodola* (48) or the approximate method of *Schwerin* (49). Theoretically, it is possible to use either flexible shafts, with their natural frequency below the speed of rotation, or rigid shafts. Both systems have their practical advantages and disadvantages.

In the case of flexible shafts, care must be taken when running through the first critical speed; moreover larger clearances between rotor and casing causing greater leakage losses are necessary on account of the greater deflection of the shaft. On the other hand, the glands become smaller, due to the smaller diameters, and the thin shaft will warm up quicker than a heavy, rigid shaft. With the latter small clearances may be employed in the blading; the glands will have larger diameters, however, and their losses will be higher; the discs will have larger bores and the stresses in the hubs will increase. The present tendency is to use rigid shafts as much as possible, the natural frequency being at least 15% above the running speed. The large diameters of the glands are reduced as much as possible by taking the right profile of shaft to get a good elastic line.

Torsional vibrations are generally of less importance in modern turbines. There are usually no impulses to produce them, except in the case of gear drives when inaccuracies may occur in the teeth. In this case also, resonance between any natural torsional frequency and the speed of rotation should be avoided. The stiffness of the shaft may be altered by increasing or decreasing the mass. The natural frequency should be at least 15% above the speed of rotation. If this is not possible it may be reduced to 15% below the running speed by increasing the momentum of the gear wheel, for instance. The arrangement to be adopted must always be considered when designing the shaft. Subsequent alterations are frequently very awkward on account of lack of space or other constructional reasons.

---

(47) Schweizerische Bauzeitung 85 (1925) p. 265.
(48) 6th (German) Edition p. 942; English Edition p. 1140.
(49) "Die Hütte", 25th Edition, Vol. 1 p. 404 (Berlin: W. Ernst & Sohn 1925).

III. Principles of design

In accordance with what has been stated in the previous pages, certain rules may be given for the calculation and design of turbines. *Most important of all are the conditions of absolute reliability and simplicity of design.* The economy or, in a narrower sense, the turbine efficiency, will only occupy the second place and will vary in accordance with the capital expenditure which the situation in the money market will justify. Reliability demands, in the first place, the adoption of moderate peripheral velocities. In this way, the stresses in the rotating parts will be kept within safe limits and the conditions to be specified for the properties of the material and the manufacture will not be unreasonable. The increase of the maximum output demanded solely to satisfy commercial competition is intolerable. The price per kilowatt and the weight may be reduced by increasing the diameters of the last stages; the stresses are, naturally, increased and care must be taken that they do not exceed the limit which experience has demonstrated as safe.

When calculating the internal design of a turbine the starting point is always the volume flowing through, and not the quantity of steam or the output. In the first place, the lengths of the blades must be sufficient and the leakage loss must be kept within reasonable limits. When the steam volume is small higher speeds will be used, irrespective of whether the turbine type is condensing, back-pressure or mixed-pressure. In this way, the blade heights will be increased. The pressure after the first stage should not be chosen too high. Turbines of medium outputs and for high steam pressures may require blade heights which are less than the practical limit, even if small diameters are used. A reaction blading would need very small blade clearances to reduce the tip loss, and the design might be unsafe. There is, therefore, a lower limit to the application of reaction blading in the H.P. part. Except for very large volumes of steam, the impulse system should always be used for the H.P. part. With its larger clearances the turbine is more reliable and will maintain its efficiency better. If reaction is to be employed from a lower pressure, the question arises, what should this pressure be. It depends especially on the leakage loss and is determined, therefore, mainly by the volume of steam. If this is large, long blades will be obtained, the clearance loss will be small and a reaction blading may be adopted from a relatively high pressure.

Concerning condensing turbines for small output or high pressure, best conditions are obtained with the impulse principle and a small number of stages. Up to about 1000 kw. it is advisable to use speeds higher than 3000 R.P.M. the right value depending chiefly on the volume of steam. The best speed for turbines of 300 to 400 kw. and for normal pressures is between 5000 and 7000 R.P.M., for smaller units it is from 7000 to 10,000 R.P.M.. For larger outputs a greater number of stages is advisable and only impulse bladings should be used in the H.P. part. Reaction will be employed in the L.P. part with stages on drums or on a small number of discs. The H.P. stages should all have a single row of moving blades if the steam quantity is large and the pressure normal. For an average flow of steam and a high pressure it is advisable to take for the first stage a velocity wheel with two rows of blades, using about one quarter to an eighth of the total heat drop.

The most noticeable feature distinguishing a back-pressure from a condensing turbine is the complete absence of the L.P. part. In back-pressure turbines also, great differences are found in the heat drops, which may be between about 55 and 325 B. Th. U./lb. (30 to 180 kcal./kg.) and may vary, thus, by as much as 500%. Condensing turbines are usually designed now for heat drops between about 340 to 520 B. Th. U./lb. (190 to 290 kcal./kg.) extreme values being only about 50% different. If it is considered what a great number of variations may occur in the steam quantity, the live steam pressure and temperature and the back-pressure it will be understood that general rules are much easier to state for high-pressure than for back-pressure turbines. In the latter case precise rules are very difficult to establish, also they are often not desired as otherwise the flexibility of design might be impaired.

When great importance is not attached to obtaining the best possible efficiency a back-pressure turbine with only a two-row velocity stage may be employed. If the heat drop is sufficiently small a single-row impulse stage may be used, higher speed of rotation being chosen for smaller steam quantity. Besides its extreme simplicity, this solution has the advantage that the efficiency remains almost constant at all loads. This may be of great value in certain cases. If higher efficiencies are required a larger number of stages will be necessary. In the region of high pressures the impulse system should be used as it is the only way of obtaining sufficient axial and radial blade clearances. The losses in the glands must always be considered in this case. Except for small volumes of steam, it is best to design a multi-stage back-pressure turbine for less steam than corresponds to full load. Overload nozzles will be used for full load and overload. If a reaction blading is employed, it will be preceded by one or more impulse stages for governing.

Single-casing turbines are generally used for small and medium outputs and for moderate steam pressures. For higher steam pressures a larger number of stages will be required and it will be an advantage to divide the heat drop amongst several casings. On account of the high cost, this arrangement is usually employed only for large outputs. Multi-casing turbines require also a large amount of space. It is not possible to build single-casing turbines with the number of stages necessary for obtaining the best efficiency when the heat drop is large. In the case of very high vacua, such as pressures of 0.6 lb./sq. in. (0.04 kg./cm.^{2}) absolute or less, it will not be possible, even for turbines of medium output, to obtain a sufficiently large blade area in the last wheel and the steam will have to be divided into two flows in the last stages. The number of casings must be kept as low as possible so as not to increase the overall length and the space required or the number of journal or thrust bearings, couplings, glands, or pipe lines which complicate the turbine and increase the price.

Another point to be remembered when designing a turbine is that the surest way of obtaining smooth running is to use rigid shafts. If flexible shafts are employed it should only be for moderate temperatures or small sizes. The material is then most likely to be homogeneous and the heat expansions uniform. It is possible to avoid vibrations in the rotors by using either wide discs of small diameter or drums. The moving blades must be fixed in such a way that there is no doubt about them being held rigidly even when subjected to the centrifugal force and heat expansions. The vibration of blades depends, to a certain extent, on the fixation. If the blades are not held firmly, the periods of vibration will depend, therefore, on the speed of rotation and it will be difficult to choose the blade shape so that synchronism never occurs at any of the running speeds. Periodic impulses, no matter what their kind, should be carefully avoided in turbines.

It must be mentioned, finally, that the tendency of using a large number of stages for the purpose of improving the efficiency is only justified if the shops

are in a position to work to absolute accuracy. Multi-stage turbines, with their great number of parts, are naturally more liable to faulty workmanship. As the output decreases or as a higher efficiency is required, the accuracy of manufacture must increase.

These fundamental principles are observed by nearly all the leading firms in their latest designs as may be seen by the examples in the following pages. Very often turbines of different manufacture only differ on a few details. The design of economical and reliable turbines is now, so to speak, common knowledge. Contradictory opinions are not often heard and when they are upheld, it is usually by those who, having started in another direction, have reasons to adhere to it.

IV. Practical applications of the principles of design

We have discussed how the economy of steam turbines may be raised, how the practical limits may be determined and how it is possible to deduce principles which must be observed in the design of modern turbines. Practical application of these principles will now be given.

1. Turbines for direct drive

As has already been stated, small capacity turbines usually run at high speeds. They will be dealt with in the second part of this chapter, which is devoted to turbines for indirect drive. The turbines described in this section are mostly for driving two or four-pole alternators. Machines made in Europe usually run at 3000 or 1500 R.P.M. for producing fifty-cycle current; the American machines are for sixty-cycles and run at 3600 or 1800 R.P.M.. These speeds have become standard except in a few cases, such as for ship propulsion or for driving compressors or large rotary pumps.

In Europe the demand is for light and cheap machines, and the higher speed is used for much larger outputs than in America, where no competition exists for raising the outputs at 3000 or 3600 R.P.M. (50). The highest capacity for a 3600 R.P.M. machine is about 15,000 kw. in America, an output which is insignificant in large electrical concerns. In Europe, on the other hand, the limit capacity for 3000 R.P.M. turbines has been increased, as has already been mentioned, up to about 50,000 kw. and high-speed machines play a great part in extensive power production.

In the following pages, 3000 R.P.M., then 1500 R.P.M. high-pressure turbines will be treated, afterwards back-pressure, extraction, low-pressure, mixed-pressure and special turbines, such as the *Ljungström* turbine, of the latest and most important types will be described.

a. High-pressure turbines

As a result of the foregoing and because large units are commonly used, it is not surprising that in America 3600 R.P.M. high-pressure turbines are regarded rather as small units. This does not mean that less care is taken over their design, but the tendency is to manufacture them so that only small alterations are necessary for adapting them to different requirements and steam conditions. The first example is a 1500 kw. impulse turbine of *Elliott* (Fig. 120). It works with steam at about 300 lb./sq. in. (21 kg./cm.^{2}) gauge, 610° F. (320° C.) at the throttle valve and 1.15 lb./sq. in. (0.08 kg./cm.^{2}) absolute back-pressure. The efficiency is given as 76% at the coupling for these conditions. An original feature is the use of glands of the labyrinth type in the H.P. region and of the carbon type in the L.P. region.

The impulse turbine of the *Amer. G.E.C.* shown in Fig. 121 has many similar points. In order to limit the number of patterns the casing is formed out of several rings with flanges of standard diameters. By combining suitable rings it is possible to build up high-pressure turbines for different steam conditions; extraction or back-pressure turbines may also be made with the same patterns. The discs are forced directly on to the shaft and the size of the hubs depends on

(50) See *F. Hodgkinson:* "Steam turbine and condensing equipment", The Electrical Journal 21 (1924) p. 508 and *O. F. Junggren:* "Large Curtis turbines", Power 61 (1925) p. 290.

the diameters of the wheels and the stresses, even parallel discs being used for the fifth and sixth stages. In order to allow for the widening of the steam jet, conical coverbands are used throughout, in spite of the increase in cost. The barrel which carries the last five diaphragms projects far into the exhaust casing and no diffuser effect can be obtained.

Larger and more efficient turbines for 3000 R.P.M. are built by all turbine manufacturers in Europe. Such a machine is the single-casing impulse turbine of the *Zoelly* type shown in Fig. 122. It is for 12,000 kw. and runs at 3000 R.P.M.; the live steam conditions are 270 lb./sq. in. (19 kg./cm.^{2}) gauge and 715° F.

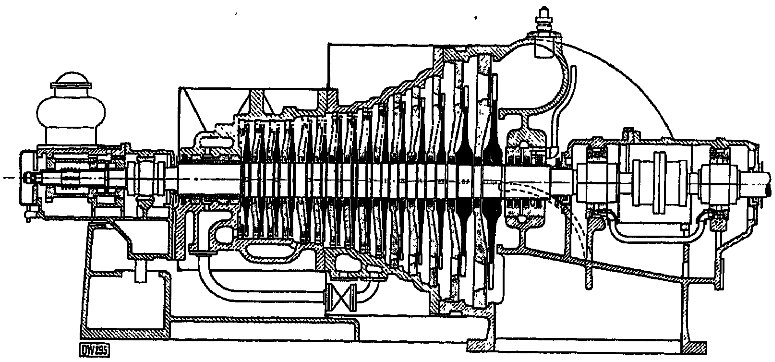


Fig. 120. *Elliott,* 1500 kw., 3600 R.P.M. high-pressure turbine

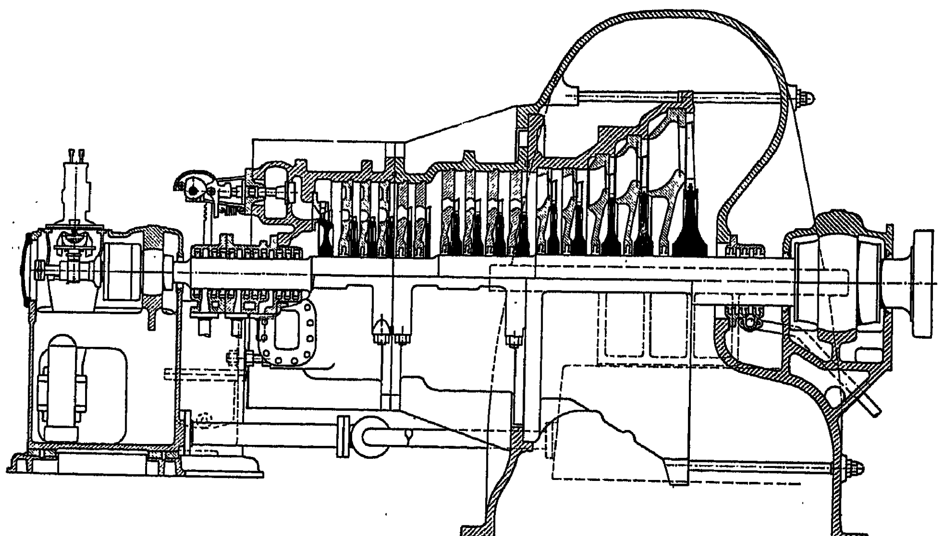


Fig. 121. *Amer. G.E.C.,* 3000 kw., 3600 R.P.M. high-pressure turbine

(380° C.) and the pressure in the condenser 0.74 lb./sq. in. (0.052 kg./cm.^{2}) absolute. The diameters increase up to 69 in. (1750 mm.) in ten stages. The machine was tested by *Dresden* (51) and a thermodynamic efficiency of 82.6% was obtained at full load. The quality figure is 14,350 $\frac{\text{ft.}^2/\text{sec.}^2}{\text{B.Th.U./lb.}}$ $\left(2400\,\frac{\text{m.}^2/\text{sec.}^2}{\text{kcal./kg.}}\right)$. A turbine of similar design and the same number of stages, but with a smaller diameter for the last stage, has been tested by *Stodola* (52) and similar results were obtained.

(51) Schweizerische Bauzeitung 91 (1928) p. 181.
(52) Z. VdI. 71 (1927) p. 747.

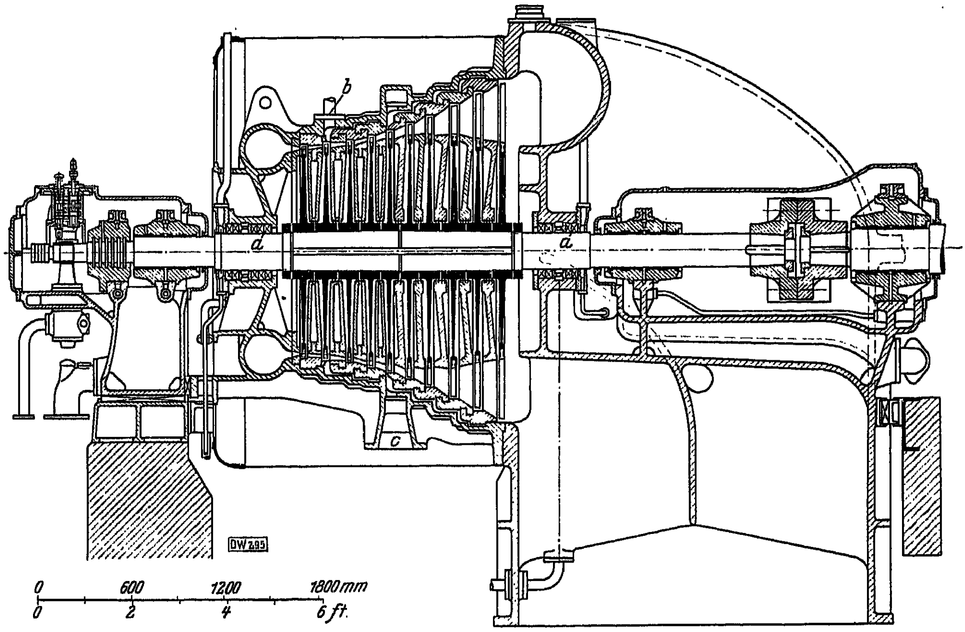


Fig. 122. *Escher Wyss,* 12,000 kw., 3000 R.P.M. high-pressure turbine.

a = Carbon-type packing gland b = Overload steam c = Bleeder branch for feed-heating

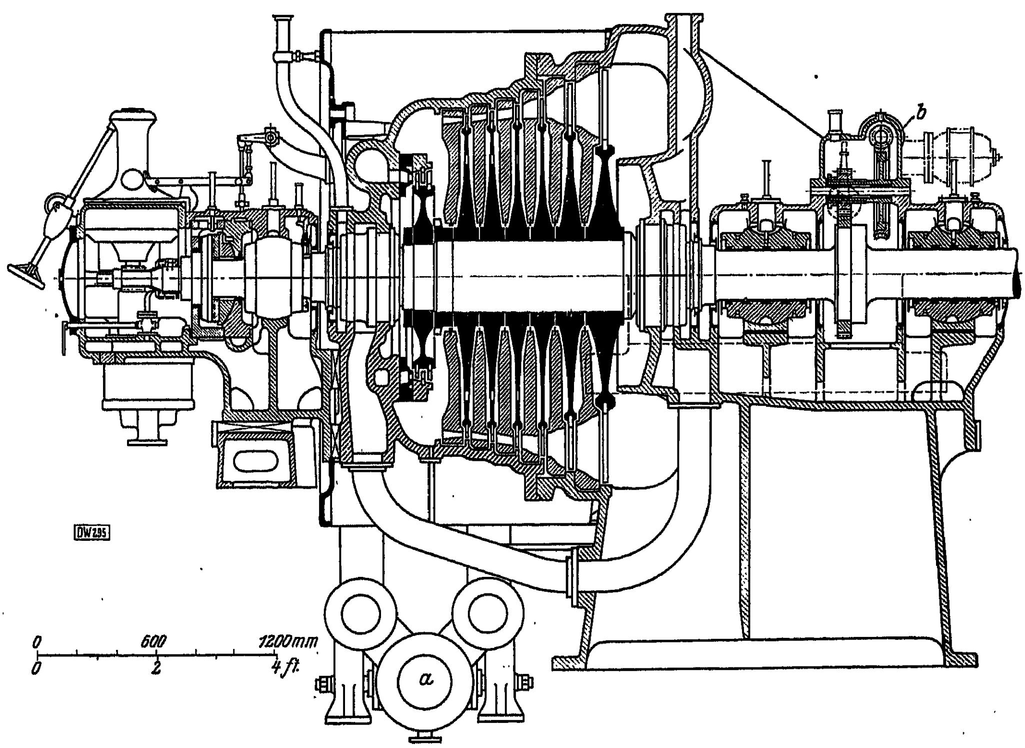


Fig. 123. *A.E.G.,* 17,500 kw., 3000 R.P.M. high-pressure turbine

a = Steam strainer b = Barring gear

The 17,500 kw. single-casing *A.E.G.* turbine shown in Fig. 123 has a similar general appearance. The first stage is a two-row velocity wheel with nozzle governing and a relatively good efficiency will be obtained at partial loads. This advantage, together with the possibility of a quick starting up of a short machine, compensates partly for the higher efficiencies which might be obtained by a turbine having more stages and casings. Consequently, such single-casing machines are particularly suited for power stations with a highly fluctuating load or for use as peak load machines. The last stage of this turbine has also a mean diameter of 69 in. (1750 mm.) and it is designed for about 30% reaction. The working conditions are 200 lb./sq. in. (14 kg./cm.^{2}) gauge, 660° F. (350° C.) and about 1 lb./sq. in. (0.07 kg./cm.^{2}) absolute back-pressure. The quality figure is about 13,750 (2300). Attention may be drawn to the barring gear; it is an accessory which is becoming more and more common for large high-pressure turbines. It shortens the time required for starting up and prevents the rotor getting bent when out of operation.

The *Parsons* pure reaction turbine in Fig. 124 is a very different kind of machine. It has throttle governing similar to the turbine shown in Fig. 122, the

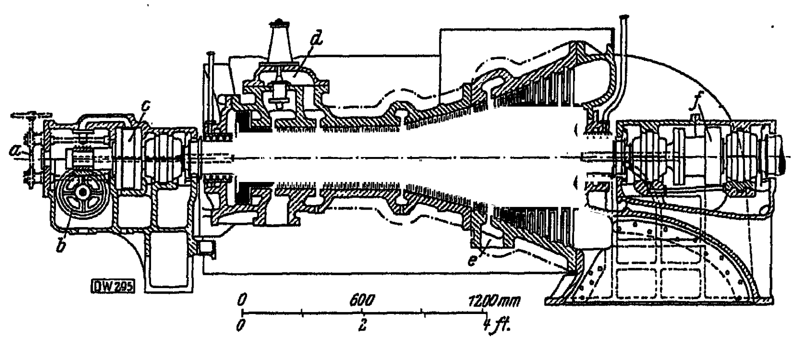


Fig. 124. *Parsons,* 3000 kw., 3000 R.P.M. high-pressure turbine

a = Thrust block adjusting gear
b = Governor and oil pump drive
c = *Michell* thrust block
d = Overload valve
e = Bleeder branch for feed-heating
f = Claw-type coupling

overload steam being admitted after the ninth stage. The economical load is 3000 kw., and 3750 kw. may be produced with a higher steam consumption. The rotor consists of a solid drum and there are no less than 55 rows of moving blades in five groups. The blade height of the first row is 0.63 in. (16 mm.) with a mean diameter of 14.2 in. (360 mm.), the last blade is 9.5 in. (240 mm.) long on 37.8 in. (960 mm.) diameter. All blades are of rustless iron. The six last rows are sealed in the radial direction, the others have the usual end tightening which has already been described (Fig. 42 *a*). Steam at about atmospheric pressure may be bled for feed-heating. The machine has carbon packing glands. Another feature is the *Michell* thrust block with the device for adjusting the position of the rotor. A similar one has already been shown in Fig. 78. The coupling between the turbine and generator is of the claw type.

The 10,000 kw. *Westinghouse* turbine in Fig. 125 is one of the largest 3600 R. P. M. American machines. It has a shorter overall length than the *Parsons* turbine as a velocity wheel is used for the first stage and the diameters of the reaction stages are larger. It is designed to be manufactured in series and may be adapted to different steam conditions by changing the blading. Four branches are provided for feed-heating.

Another reaction turbine of the *Parsons-B. B. C.* type is shown in Fig. 126. It is for outputs up to 10,000 kw.. In this example *B.B.C.* have abandoned the

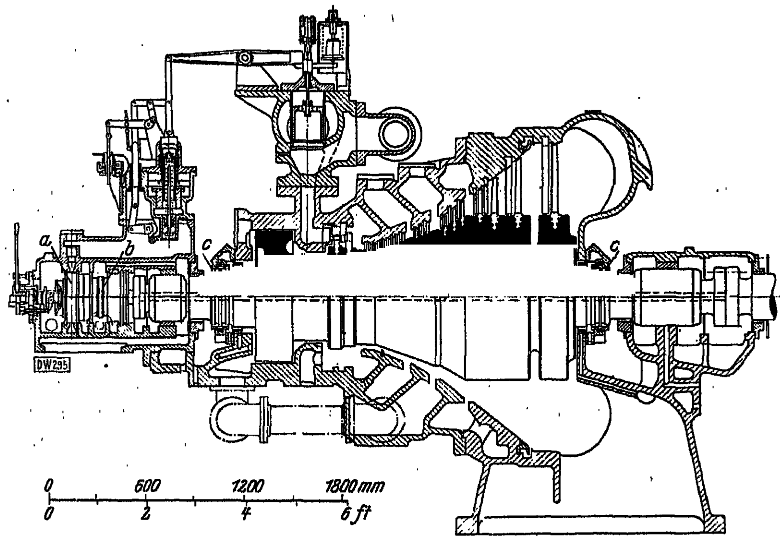


Fig. 125. *Westinghouse,* 10,000 kw., 3600 R.P.M. high-pressure turbine
a = Governor oil pump b = Pad-type thrust block c = Water-sealed gland

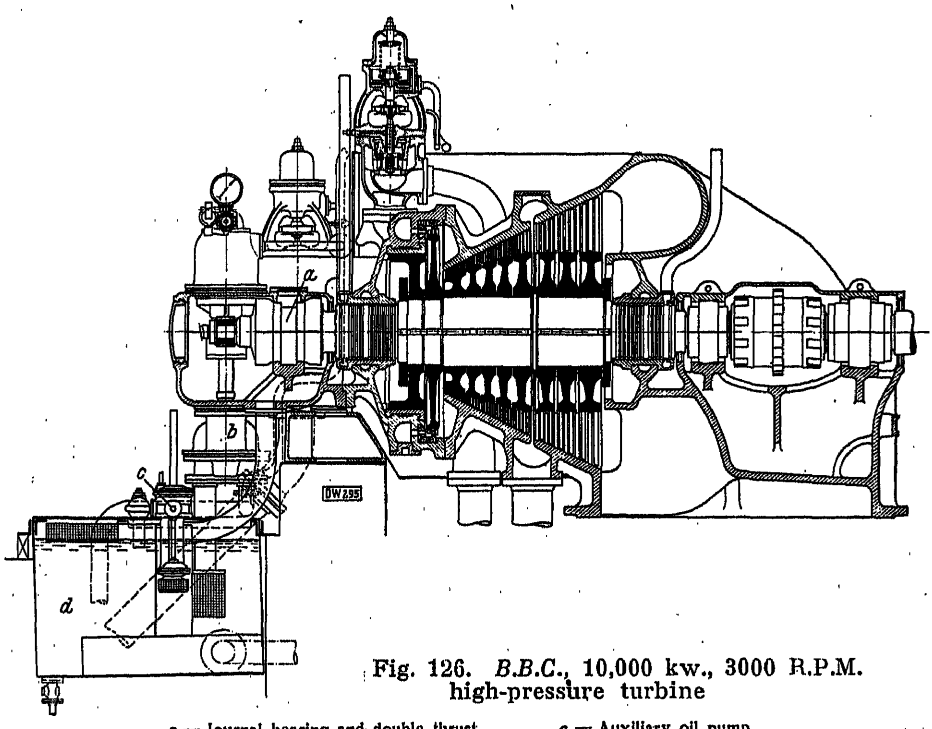


Fig. 126. *B.B.C.,* 10,000 kw., 3000 R.P.M. high-pressure turbine

a = Journal bearing and double thrust bearing with movable pads
b = Main oil pump
c = Auxiliary oil pump
d = Oil tank

original form of drum, usually consisting of two forgings, and the reaction stages are mounted in groups of three or four on separate discs. In this way, warming up is quicker and more uniform, and the turbine may be more readily started up (53). A characteristic feature which may be noted is the large

(53) See *F. Ribary:* "Verkürzung der Anfahrzeit bei Dampfturbinen", p. 510, "Festschrift Stodola" (Zürich: Orell Füssli 1929).

difference between the diameters of the regulating stage and the first reaction stage with full admission. The remainder of the blading widens out continuously and smoothly up to the last stage. Branches for feed-heating are provided also in this machine. The disc at middle of the coupling has teeth cut out of it and the rotor may be turned by hand with a ratchet.

Even *Parsons*, the inventor of the reaction turbine, uses impulse blading for small steam volumes. In Fig. 127 an example is shown. It is a combined impulse-reaction turbine of 1500 kw.. There is a two-row velocity wheel followed by nine single-row impulse stages and a reaction drum with eight rows of moving blades. The steam conditions being about 250 lb./sq. in. (17.5 kg./cm.²) gauge, 700° F. (375° C.) and 95% vacuum, the quality figure is only 6575 (1100). It is, unfortunately, not known what turbine efficiency has been obtained. The blade velocities are exceedingly small, the mean value for the last row being only about 416 ft./sec. (127 m./sec.). The rotor has been machined out of a solid forging and has been bored through to allow the metal at the centre to be examined. The first three diaphragms have built up nozzles, the remainder are cast in. The other details of the turbine are similar to those of the machine

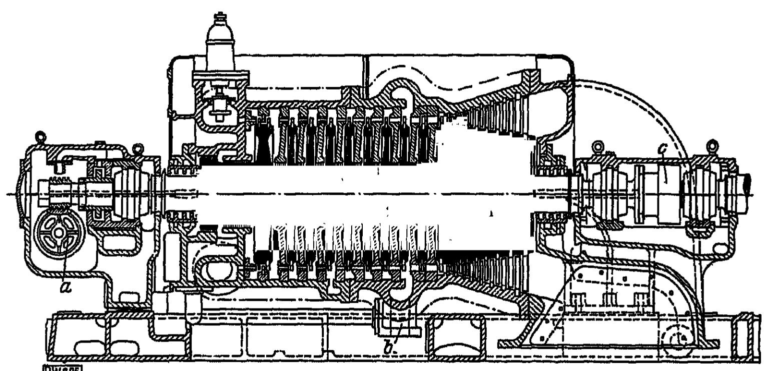


Fig. 127. *Parsons,* 1500 kw., 3000 R.P.M. high-pressure turbine
a = Governor and oil pump drive b = Bleeder branch for feed-heating c = Claw-type coupling

shown in Fig. 124. Provision is made, in this case also, for bleeding steam at about 15.7 lb./sq. in. (1.1 kg./cm.²) absolute for feed-heating. It will be seen later (page 158) by a further example that there are other *Parsons* designs with impulse wheels for small steam volumes.

This type of *Parsons* turbine is very similar to the *A.E.G.* high-pressure turbines for small outputs. Fig. 128 shows a machine of this kind for 2200 kw. at 3000 R.P.M.. It has a two-row velocity stage for small heat drop, a number of impulse wheels of slightly increasing diameters and some reaction stages on a drum. The Σu^2 is 4,030,000 ft.²/sec.² (375,000 m.²/sec.²) and the quality figure about 9260 (1550), values which are higher than for the *Parsons* turbine on account of larger diameters being employed throughout. For suitable steam conditions the machine has an efficiency of about 78% at the coupling. The two-row velocity wheel enables the turbine to adapt itself to varying conditions. A feature of this design is the large amount of overload which may be obtained, the turbine being able to give full load with greatly reduced steam pressure.

Erste Brünner have also returned of late to a design with less stages. Thus, the 10,000 kw. high-pressure turbine in Fig. 129 resembles the machine in Fig. 128 except for certain details in construction.

The longest permissible blade determines the permissible leaving area and, for every back-pressure, it limits the output of a single-flow machine for

a given speed. For larger outputs either a lower speed must be chosen or the steam must be divided into several flows, each expanding in a separate part of turbine down to the back-pressure. This idea was first realized by the *Brit. G.E.C.* and *Metro-Vick* in different ways (see Fig. 54 *d* and *c*). *Parsons* recently adopted the first named arrangement. The turbine in Fig. 130

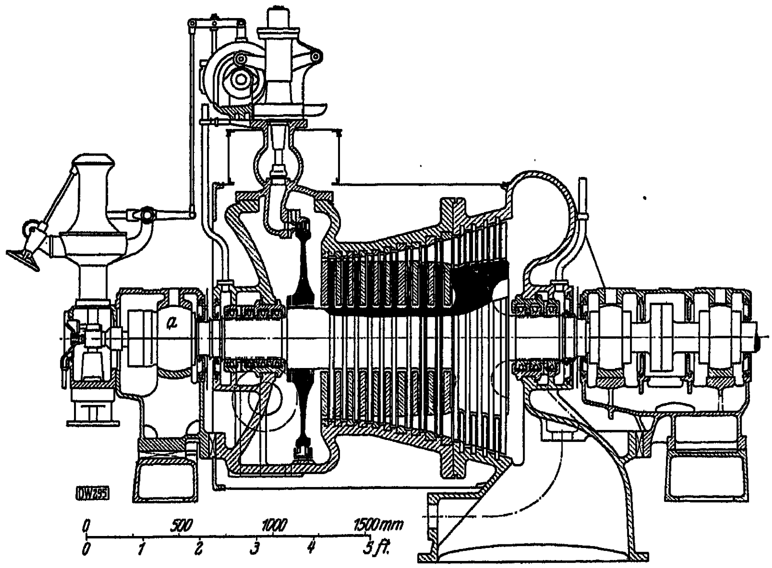


Fig. 128. *A.E.G.*, **2200 kw., 3000 R.P.M. high-pressure turbine**
a = Pad-type thrust bearing combined with journal bearing]

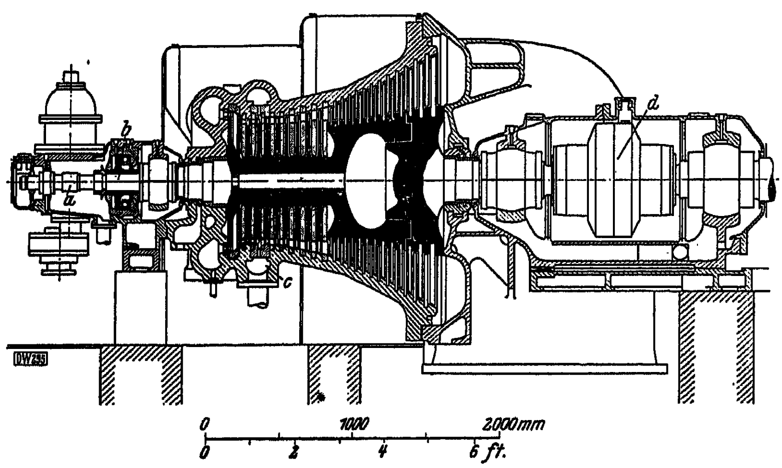


Fig. 129. *Erste Brünner*, **10,000 kw., 3000 R.P.M. high-pressure turbine**

a = Governor drive
b = Thrust bearing with pads resting on balls
c = By-pass belt
d = Double claw-type coupling

gives 12,000 kw. at 3000 R.P.M. and similar machines are built up to 20,000 kw. for about 0.5 lb./sq. in. (0.035 kg./cm.^{2}) absolute condenser pressure.

Metro-Vick employ the system of multiple exhausts as developed by their Chief Engineer, *Baumann*, as has already been mentioned (Fig. 54 *c*). An interesting example is shown in Fig. 131. This turbine is also designed for 0.5 lb./sq. in. (0.035 kg./cm.^{2}) absolute condenser pressure. An original method

has been adopted for securing the blades to the shaft. The ordinary disc design has been abandoned, obviously on account of the unusually heavy blades. Discs in halves have been used instead. They are provided with grooves to fit into the shaft and are held in place by bolts parallel to the axis.

The simplest way of dividing the steam flow is to separate the turbine into several casings. After the H.P. part the steam can either expand symmetri-

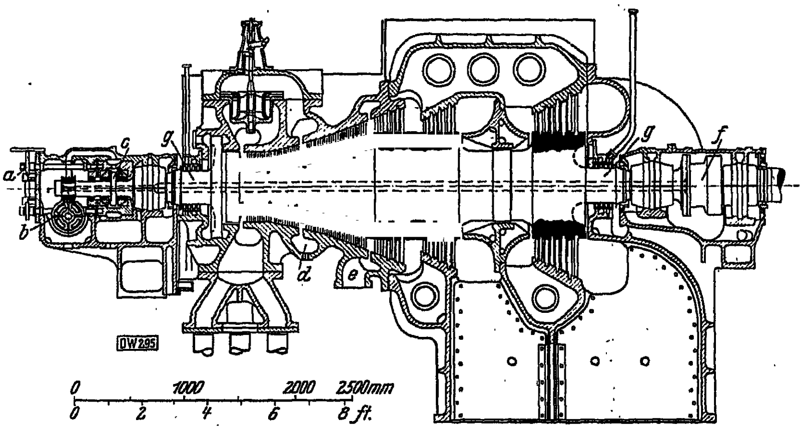


Fig. 130. *Parsons*, 12,000 kw., 3000 R.P.M., single-casing high-pressure turbine with double-flow L.P. end

a = Hand-operated thrust block adjusting gear
b = Governor drive
c = *Michell* thrust block
d = By-pass belt
e = Bleeder branch for feed-heating
f = Claw-type coupling
g = Carbon-type packing gland

cally from the middle of a double-flow L.P. casing or each flow may be taken to a separate casing. By these means the total number of stages may be raised without unduly increasing the bearing centres or the diameters of the shaft or glands. It was for this reason only that turbines were built for some time in several casings to meet the demand for a good steam consumption. Multi-casing but single-flow turbines were built for even relatively small capacities. This

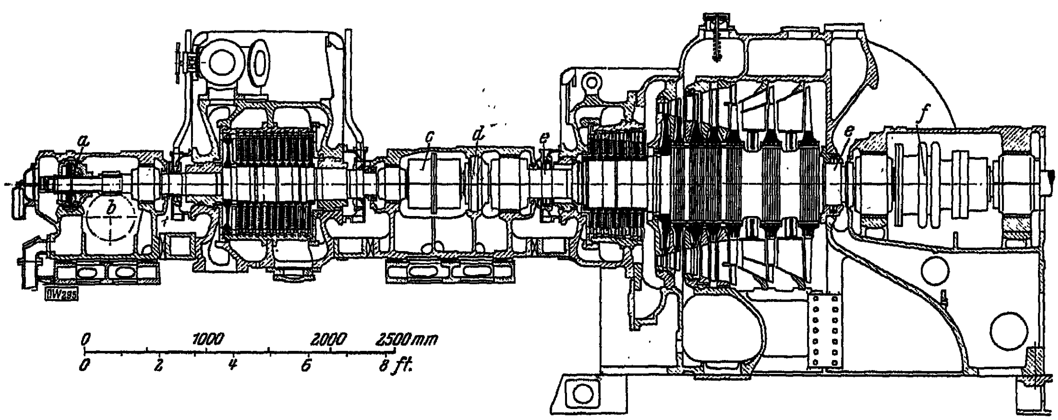


Fig. 131. *Metro-Vick*, 12,500 kw., 3000 R.P.M., two-casing high-pressure turbine with multiple exhaust

a = H.P. thrust block
b = Governor drive
c = Double claw-type coupling
d = L.P. thrust block
e = Water-sealed gland
f = Coupling of the corrugated pipe type

method has now been abandoned and, except in special cases (e. g. for high pressures), several casings are only provided when the leaving area requires the provision of two or more flows.

It is for this reason that during the last few years the double-flow turbine in two casings has become almost the standard design, for all types, in power stations in Europe. Owing to its advantages it should maintain its leading position in the near future. The H.P. casing, which is exposed to the high

pressures and temperatures of the live steam, is of relatively small dimensions, it may be of strong design and made of cast steel. The large L.P. casing, which is only subjected to low heat stresses, may be made entirely of cast iron. The individual shafts are shorter and lighter; the supports of the casings can more easily take up the heat expansions. The double-flow L.P. part is usually perfectly symmetrical, axial thrusts are avoided in impulse or reaction turbines alike, and balance pistons or heavily loaded thrust bearings are not required.

Fig. 132 shows a two-casing turbine with a double-flow L.P. part. It is of the *Zoelly* pure impulse type built by *Escher-Wyss*. The output is 27,000 kw. and the steam conditions are 355 lb./sq. in. (25 kg./cm.²) gauge, 660° F. (350° C.) at the stop valve. The details of construction are the same as shown in Fig. 122. Throttle governing is also employed in this case, the steam flow is, however, divided between two valves which are connected and open simultaneously. The

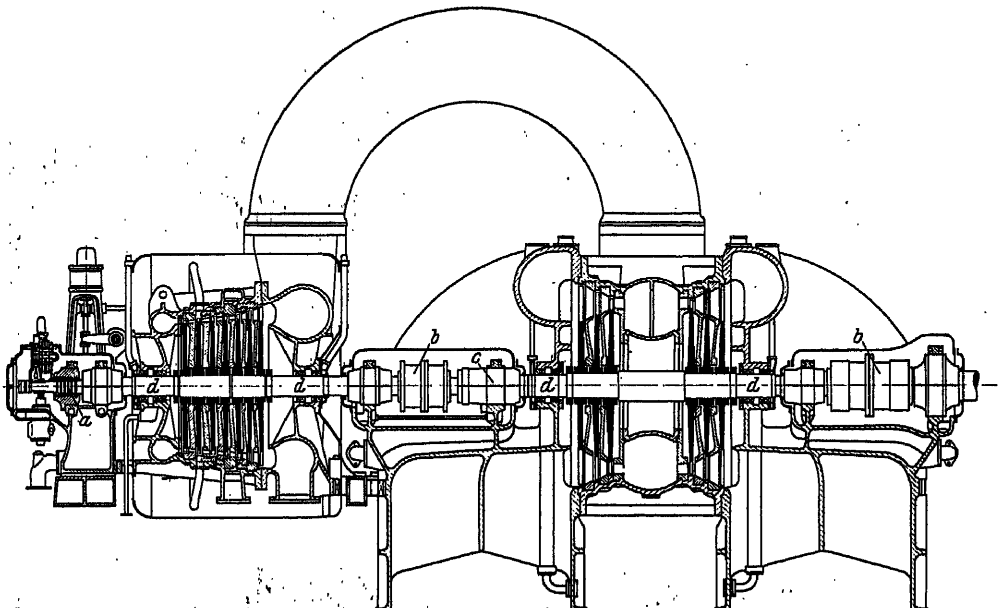


Fig. 132. *Escher Wyss,* 27,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = H.P. thrust block of the multi-collar type *c* = L.P. thrust block of the multi-collar type combined with the front L.P. journal bearing
b = Flexible coupling *d* = Carbon-type packing gland

two rotors are coupled together and to the generator by claw-type couplings and two multi-collar thrust bearings are provided. Two branches on the H.P. casing are provided for feed-heating.

Another pure impulse turbine, but of different design, is given in Fig. 133. The latest method of construction of the *Brit. G.E.C.* as developed by their Chief Engineer, *Pochobradsky,* is very well shown. The H.P. stages are all of small diameter, they all have full admission and are carried on a drum. Full admission in the first stage corresponds to the economical load, overload steam is introduced at a lower stage. The H.P. cylinder is single-flow, the steam expanding away from the centre bearing pedestal towards the front end of the turbine. The L.P. turbine is double-flow and has two sets of stages on five discs. The long blades in the last stages are reduced in width towards the tip in several steps, an arrangement which has the same purpose as the usual gradual reduction of section. The turbine produces 30,000 kw. at 3000 R.P.M.. The H.P. casing has two branches for feed-heating, the L.P. casing has a third one.

The turbine shown in Fig. 134, besides being a characteristic double-casing machine with a double-flow L.P. part, shows the methods of design of *S.S.W.-Roeder.* The original features of this turbine are the discs of large diameters with solid hubs after the H.P. drum and in the L.P. part. The large diameters which thus can be used will reduce the leaving loss. The separate parts of the rotor are spigoted and held together by fitting bolts.

The most common type of double-casing turbine has an impulse blading in the H.P. part and the entire L.P. part designed for reaction. The turbine

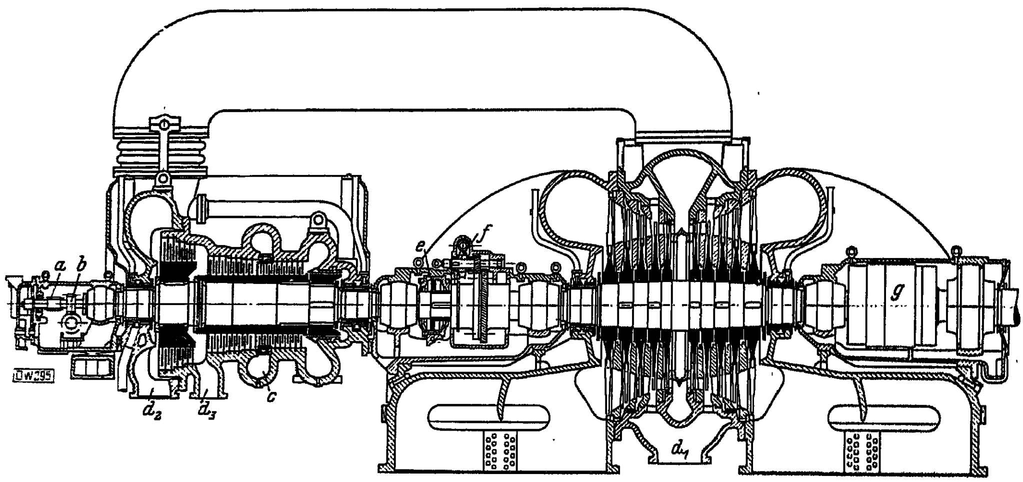


Fig. 133. *Brit. G.E.C.*, 30,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = Oil pump drive
b = Governor drive
c = By-pass belt
$d_{1 \div 3}$ = Bleeder branches for feed-heating
e = *Michell* thrust block
f = Barring gear
g = Double claw-type coupling

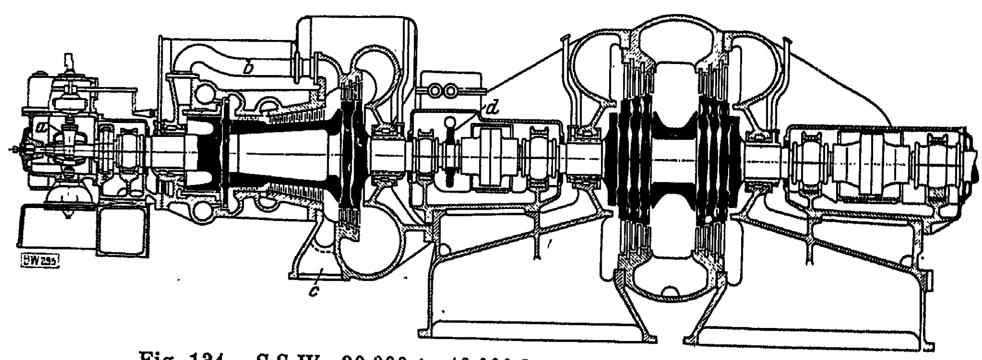


Fig. 134. *S.S.W.*, 30,000 to 40,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = Governor drive
b = Steam pipe to balance piston
c = Bleeder branches for feed-heating
d = Barring gear

of the *English Electric* in Fig. 135 is of this type. Its output is about 20,000 kw.. The H.P. rotor has 16 single-row impulse wheels which have been machined out of the solid. The double-flow L.P. rotor is of the solid drum type with a large number of reaction stages and is similar to the well-known *Parsons* double-flow machine. The steam passages in the H.P. and the L.P. parts increase very smoothly in section. The H.P. gland is of the labyrinth type, the other three have carbon packings. Each rotor has its own thrust block the one for the L.P. turbine being designed to take thrusts in either direction. It will be loaded by the turbine rotor only under exceptional conditions as there is nor-

mally no thrust from the double-flow rotor. The coupling being rigid, it also serves to keep the alternator rotor in position. The H.P. and L.P. shafts are connected by a claw-type coupling. Branches are provided in the L.P. casing for feed-heating.

The double-casing high-pressure turbines for limit capacities at 3000 R.P.M. built by *Erste Brünner* (Fig. 136) and by the *A.E.G.* are of similar appearance to the machine just described as far they have a drum-type L.P. part. They are

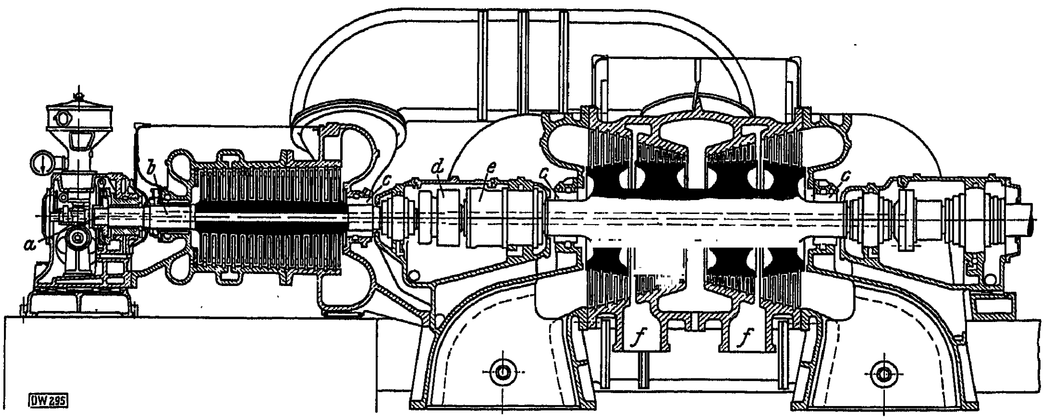


Fig. 135. *English Electric,* 25,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = H.P. thrust block of the pad type; c = Carbon-type packing gland; e = L.P. thrust block of the pad type, and journal bearing
b = Labyrinth-type packing gland; d = Toothed-type coupling; f = Bleeder branch for feed-heating

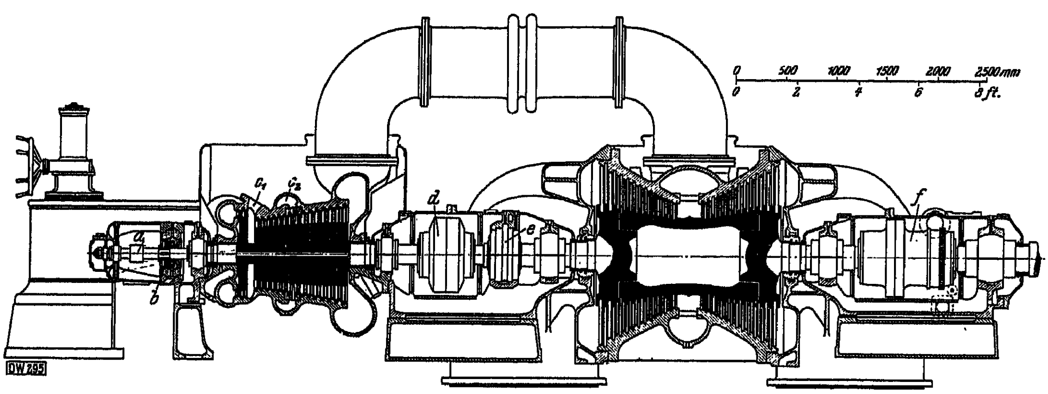


Fig. 136. *Erste Brünner,* 30,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = Governor drive; $c_{1,2}$ = By-pass belts; e = L.P. thrust bearing of the pad type
b = H.P. thrust block of the ball type; d = Double claw-type coupling; f = Double claw-type coupling with gear wheel of barring gear

always provided, however, with a stage for governing, which may be a two-row velocity wheel or an impulse stage of large diameter and with partial admission.

Discs are able to withstand higher peripheral velocities than drums and a somewhat larger leaving area is obtained when they are used for the last stages as was done in the example shown in Fig. 134. The limit capacity of a double-casing turbine of this kind, with a double-flow L.P. cylinder, is between 30,000 and 40,000 kw. for ordinary steam conditions. One of the most recent *A.E.G.* turbines may be seen in Fig. 137. The L.P. part has two sets of only three reaction stages. The quality figure of the whole turbine is about 12,550 (2100).

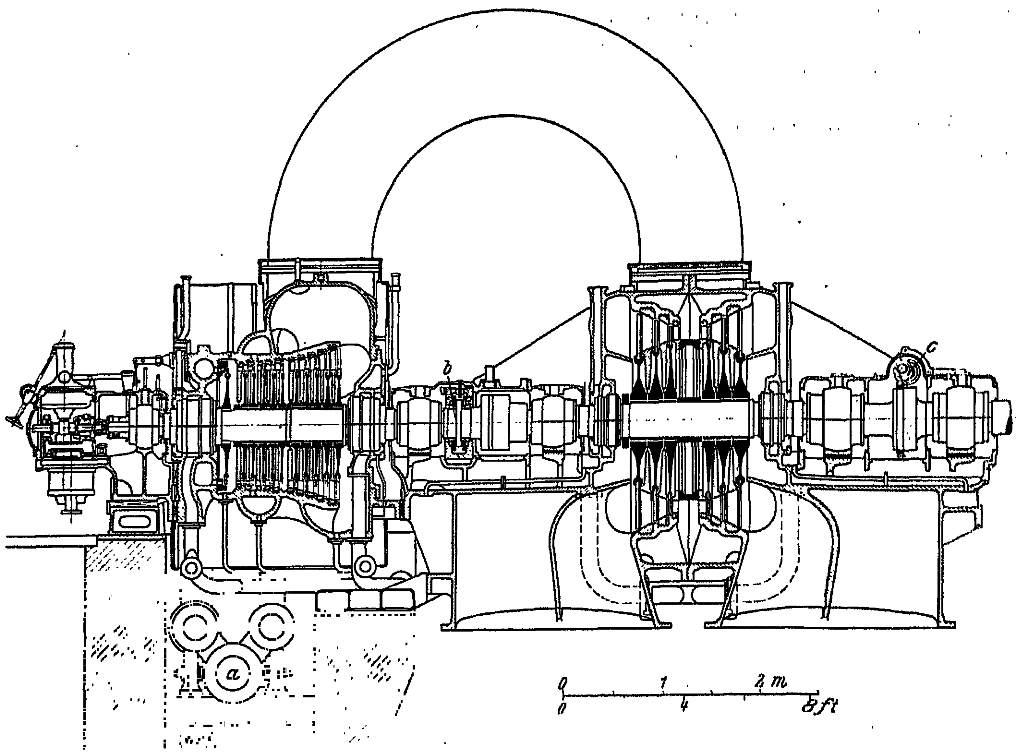


Fig. 137. *A.E.G.*, 30,000 to 40,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = Steam strainer b = *Michell* thrust bearing c = Barring gear

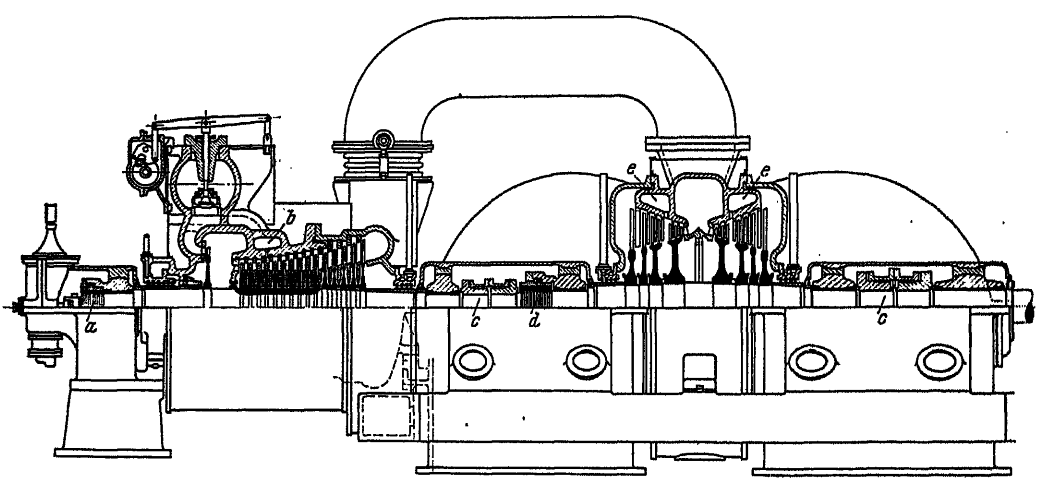


Fig. 138. *B.T.H.*, 25,000 kw., 3000 R.P.M., two-casing high-pressure turbine

a = H.P. thrust block of the multi-collar type
b = By-pass belt
c = Double claw-type coupling
d = L.P. thrust block of the multi-collar type
e = Belt for bleeding feed-heating steam

This type of machine allows larger steam quantities to be dealt with, with reasonable leaving loss, than one of the drum type; it will be quicker to start up, having a slightly smaller inertia. Which of these solutions should be adopted depends always on the local circumstances and is not a technical question. Both types have proved themselves to be thoroughly reliable for continuous service.

The 25,000 kw. *B.T.H.* turbine in Fig. 138 is of similar design. Apart from certain details such as the multi-collar thrust block, it differs from the machine just described by having two and three rows of blades on the first two L.P. wheels. It may also be noticed that the diaphragms have been omitted in the L.P. stages as is done by *B.B.C.*.

A recent design of *Parsons* has already three exhaust openings for a 25,000 kw. turbine with a pressure of about 0.5 lb./sq. in. (0.035 kg./cm.^{2}) absolute in the condenser (Fig. 139). An exhaust casing, which is the same as one half of the double-flow L.P. casing, is joined to the H.P. casing. The other exhaust casing is of the usual double-flow reaction type. The exhaust steam

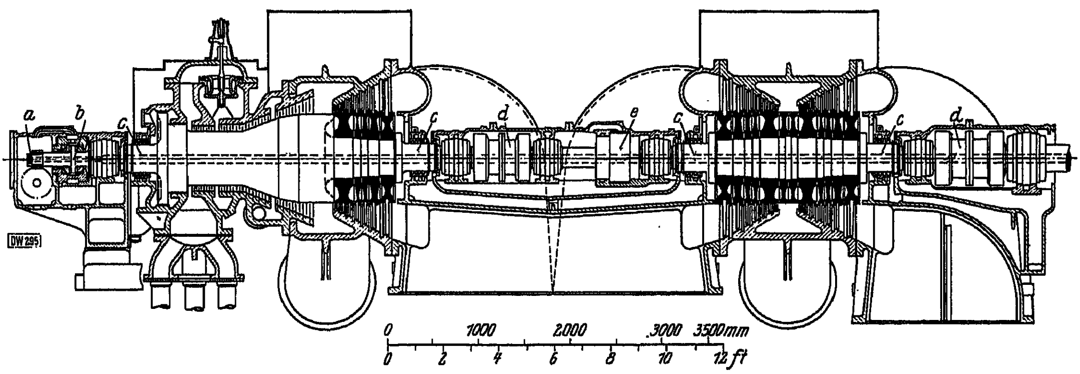


Fig. 139. *Parsons,* 25,000 kw., 3000 R.P.M., two-casing, three-flow high-pressure turbine

a = Governor drive
b = H.P. thrust block of the *Michell* type
c = Carbon-type packing gland
d = Double claw-type coupling
e = L.P. thrust block of the *Michell* type

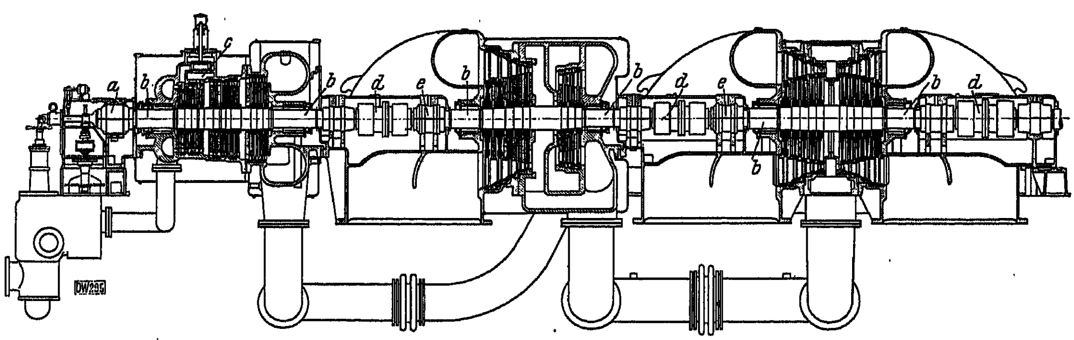


Fig. 140. *Oerlikon,* 30,000 to 50,000 kw., 3000 R.P.M., three-casing, three-flow high-pressure turbine

a = H.P. thrust block of the multi-collar type
b = Carbon-type packing gland
c = By-pass belt
d = Double claw-type coupling
e = Combined journal and thrust bearing

from the H.P. cylinder and from the first part of the L.P. casing go to the same condenser. A second condenser with half the cooling surface is connected to the third exhaust opening. The arrangement of the L.P. drum may be noted. It is divided into a series of discs as has been done recently by *B.B.C.* also.

Oerlikon have designed similar turbines with a triple-flow L.P. part. The turbine shown in Fig. 140 will give 30,000 to 50,000 kw. at 3000 R.P.M.. The main details may be seen on the sectional arrangement.

Oerlikon and *Wumag* have gone even further to increase the limit capacity at 3000 R.P.M.. They make quadruple-flow exhaust. *Oerlikon* are now building what will be the largest 3000 R.P.M. turbine. It is a four-casing turbine for 50,000 kw. on one shaft with two identical double-flow L.P. turbines. *Wumag*

arrange quadruple-exhaust in a double-casing machine (Fig. 141). The method adopted for doubling the leaving area is to direct the exhaust from two groups of stages towards each other and lead the steam away through a common opening. This arrangement has been adopted twice and there are only two flanges to connect to condensers. It is possible, therefore, to employ twin condensers placed across the axis of the turbine as is done in the case of double-flow machines. The large pipes between the two casings are certainly good from a thermodynamic point of view, they do, however, give the turbine an unusual appearance. The total number of stages between the throttle valve and condenser is relatively small. The turbine has, therefore, a low quality figure and the steam consumption is not the best obtainable. The turbine is, however, remarkable as being a successful attempt to raise the limit capacity by increasing the leaving areas. It is well suited for cases where greater importance is attached to producing the largest possible output in a small space than in obtaining a good steam consumption. Peak load stations, for instance, are designed on these lines. Details in construction which may be noted are the flexible coupling between the two turbines, the

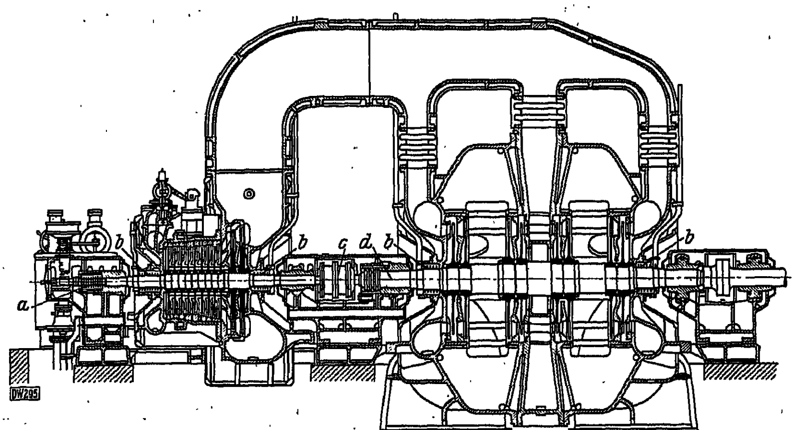


Fig. 141. *Wumag,* 33,000 kw., 3000 R.P.M., two-casing, four-flow high-pressure turbine

a = H.P. thrust block of the multi-collar type
b = Carbon-type packing gland
c = Double claw-type coupling
d = Journal bearing and L.P. thrust block of the multi-collar type

rigid coupling of the alternator and the multi-collar thrust bearing of the H.P. and L.P. rotors.

Some manufacturers design three-casing turbines with double-flow L.P. part for the range of output for which double-casing machines with double-flow L.P. part are employed. The L.P. casing will be similar to that of a double-casing turbine and the H.P. part is divided into two turbines. This will be done with impulse turbines chiefly when the first stage is not a velocity wheel. No large heat drop will then occur in the first stage and as the H.P. part in one casing would have a relatively high temperature, it is preferable to divide it into two casings of smaller dimensions. This is the principle of *Erste Brünner*'s design, a characteristic example of which is the *M.A.N.* turbine in Fig. 142. This machine is for 17,500 kw. and works with steam up to 470 lb./sq. in. (33 kg./cm.^{2}) gauge, 750° F. (400° C.). It is obvious that the large number of bearings and couplings will greatly increase the overall length and the large number of glands will occasion greater leakages. The total weight is smaller, but the price is higher, the manufacture of the small H.P. parts being expensive.

It is for similar reasons that *B.B.C.* employ three-cylinder designs, although in this case a velocity wheel or one or two single-row impulse stages are always

provided for governing. The large number of stages required for a reaction blading would unduly lengthen the rotor and the bearing centres and, in this case also, the H.P. portion of the heat drop has been divided amongst two turbines (Fig. 143). The machine shown is for 25,000 kw. and works with steam at 725 lb./sq. in. (51 kg./cm.²) gauge and 825° F. (440° C.). It was built for the well-known Belgian high pressure power station at Langerbrugge. For reasons already stated, the drums in the first two casings are split into several discs. The H.P. and I.P. rotors are rigidly coupled and are held in place by a common thrust block, which is combined with the centre bearing. The L.P.

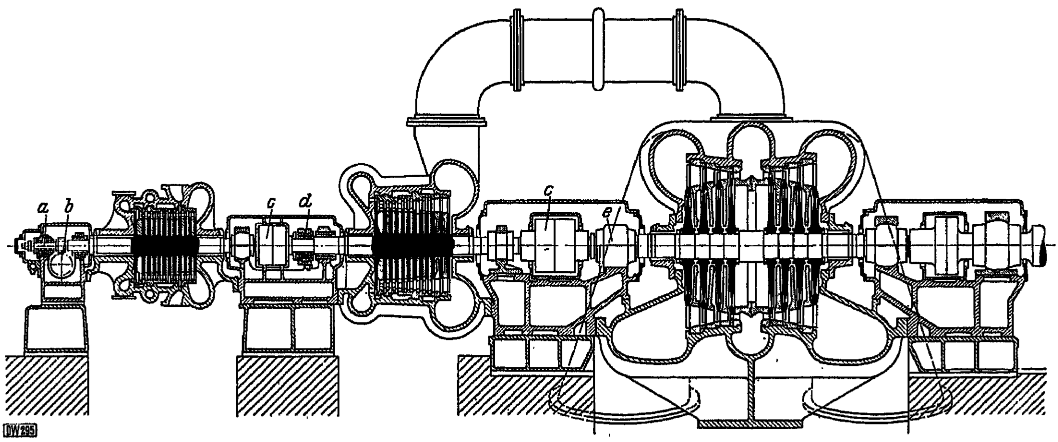


Fig. 142. *M.A.N.*, 17,500 kw., 3000 R.P.M., three-casing, double-flow high-pressure turbine

a = H.P. thrust block of the pad type
b = Governor drive
c = Double claw-type coupling
d = I.P. thrust block of the multi-collar type
e = Journal bearing combined with the L.P. thrust block of the pad type

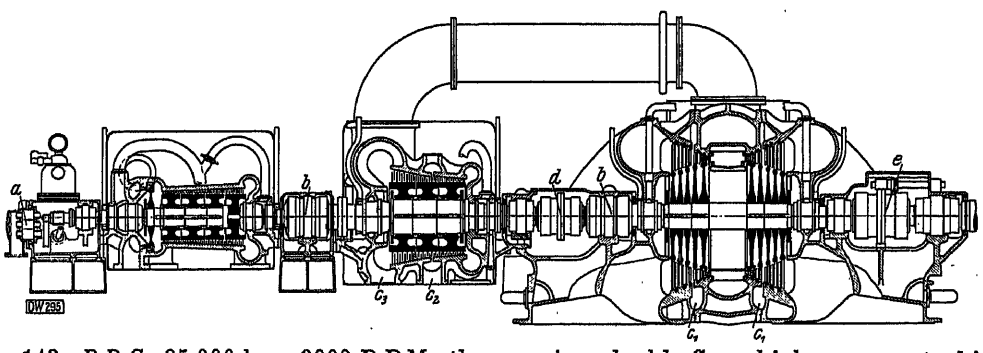


Fig. 143. *B.B.C.*, 25,000 kw., 3000 R.P.M., three-casing, double-flow high-pressure turbine

a = Oil pumps
b = Journal bearing and double thrust block of the pad type
c 1÷3 = Bleeder branches for feed-heating
d = Double claw-type coupling
e = Double claw-type coupling with barring gear

rotor has its own thrust block. The balance piston, which was formerly required in reaction turbines to equalize the axial thrust of the blades, is not necessary as the two first casings have been built for flows in opposite directions and balance each other. In spite of its great length the turbine has a compact and neat appearance. All the latest turbines are tending to abandon ungainly and often massive forms, and even when they occupy a large space, as in this case, they are of a light and pleasing design and represent a sound technical solution.

Sometimes the number of stages has been increased still further and even for moderate outputs four-casing turbines have been built having three expansions and two L.P. parts in parallel. An example is given in Fig. 144.

All four turbines have the same centre line and drive one generator. The set was built by *Stork* from *Erste Brünner*'s designs. It gives 16,000 kw. at 3000 R.P.M.. The Σu^2 is about 8,065,000 ft.^{2}/sec.^{2} (750,000 m.^{2}/sec.^{2}), and the average quality figure about 17,325 (2900). The turbine is for steam at 455 lb./sq. in. (32 kg./cm.^{2}) gauge and 750° F. (400° C.). At the acceptance tests (54) an efficiency of 82.9% at the coupling was obtained under conditions closely approaching those specified.

With multi-casing designs having small diameters and low peripheral speeds, there is no difficulty in using the number of stages and blade lengths giving the best flow. Consequently, it is possible to obtain for each separate turbine small pressure and temperature drops, short bearing centres, small, sound and easily annealed castings, greatly reduced heat expansions and distortions when in service and, in general, very moderate stresses. Against these undisputable advantages of multi-casing designs the following unavoidable disadvantages must be noted: the great number of journal and thrust bearings, of glands and of couplings, the large and long piping between the casings, which give rise to radiation losses and, together with the many oil, packing steam and drain pipes, hamper the accessibility of the turbine. The overall length will be increased also and probably the cost of the building will rise; for the same expense of materials the foundations will be less solid; lastly, the maintenance and initial cost will be high. If about the same efficiency can be obtained with turbines of different numbers of casings, preference should be given to the solution with the smallest number.

If the required output cannot be obtained with a double-flow L.P. end at 3000 R.P.M. the speed of rotation should be decreased. In Europe 1500 R.P.M. will be chosen, in America 1800 R.P.M. or, as was formerly common practice, 1200 R.P.M.. On account of the requirements of power stations the construction of large turbines for high speeds is more advanced in Europe. The Americans have the lead in the realm of slow speed turbines, of which they possess the greatest number and the largest units, in

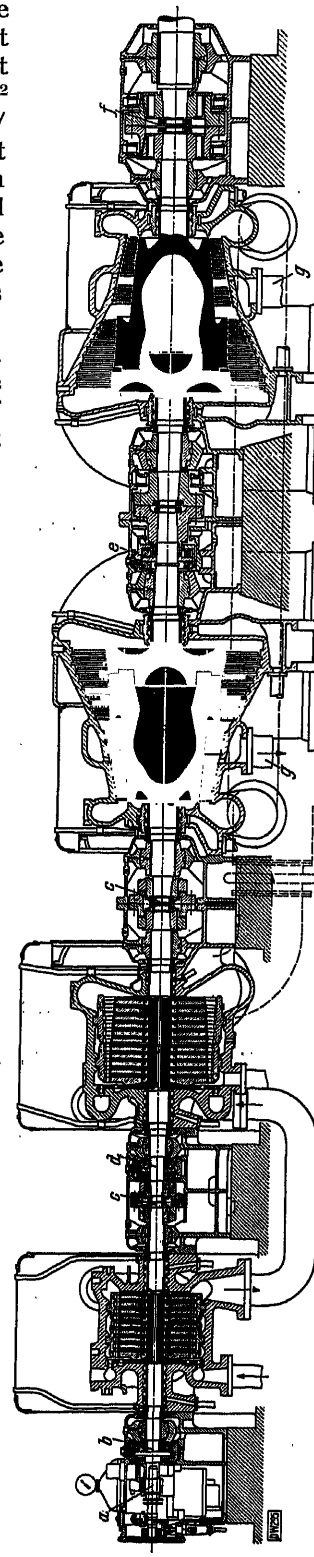


Fig. 144. *Stork*, 16,000 kw., 3000 R.P.M., four-casing, double-flow high-pressure turbine

a = Governor drive
b = H.P. thrust block of the pad type
c = Double claw-type coupling
d = I.P. thrust block of the pad type
e = Thrust block for the two L.P. turbines, of the pad type
f = Double claw-type coupling
g = Bleeder branch for feed-heating

(54) See The Engineer 144 (1927) p. 58.

spite of certain notable achievements in Europe. This is particularly noticeable in connection with very large turbines of over 50,000 kw., for which, so far, there has been but very small demand in Europe. At the same time the tendency to limit the number of casings as far as possible is much stronger in America than in Europe, especially in recent times. It may almost be stated that the Americans only adopt a multi-casing design when the output cannot be dealt with in one generator. Naturally, this rule has its exceptions, it gives, however, the general tendency.

A 1800 R.P.M. plain impulse turbine of the disc type made by the *Amer. G.E.C.* is shown in Fig. 145. It is for 75,000 kw. and has 17 stages of increasing diameters, the first being a two-row velocity wheel. The Σu^2 is about

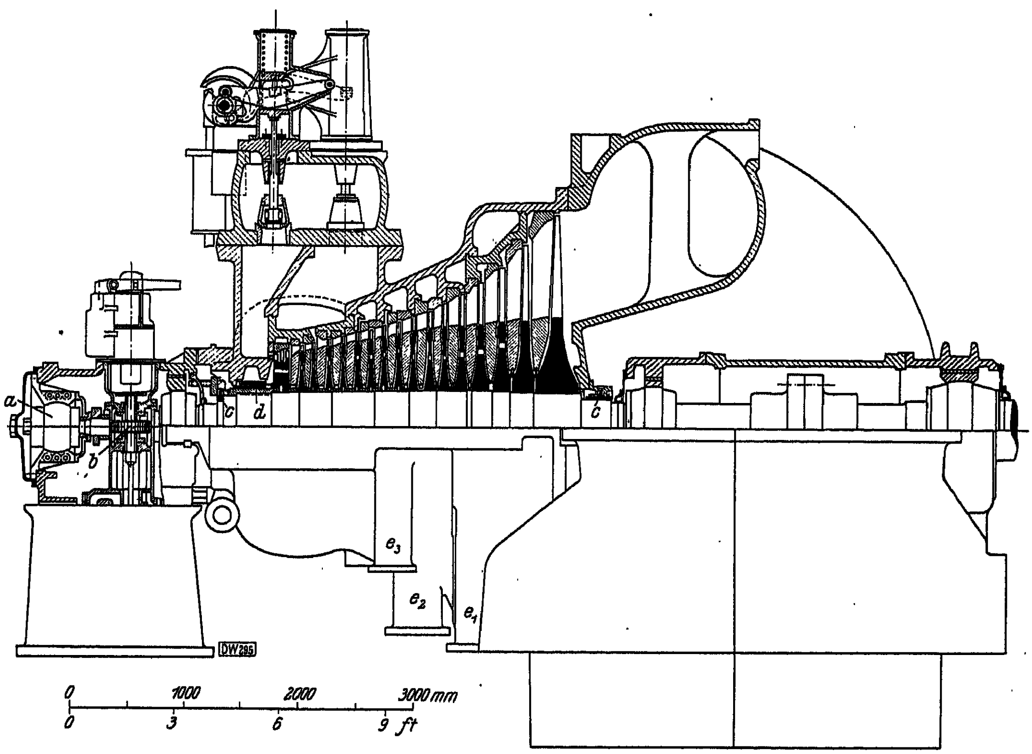


Fig. 145. *Amer. G.E.C.* 75,000 kw., 1800 R.P.M. high-pressure turbine

a = Journal bearing combined with thrust block of the multi-collar type
b = Governor and oil pump drive
c = Water-sealed gland
d = Three-flow packing gland
$e_{1 \div 3}$ = Bleeder branches for feed-heating

5,800,000 ft.²/sec.² (540,000 m.²/sec.²) and with 370 lb./sq. in. (26 kg./cm.²) gauge, about 680° F. (360° C.) and 97% vacuum — an assumption which corresponds to the average American conditions — a quality figure of about 10,150 (1700) is obtained. The efficiencies which have been published for similar machines are about 80% at the coupling, and are quite good when the poor quality figure is considered. The steam path is always very carefully designed, it increases very gradually in section and has sufficiently long blades from the beginning. The last stage has a diameter of 110 in. (2800 mm.), the disc being about 75.6 in. (1920 mm.). Naturally, the use of discs of such large diameters requires careful investigations and tests on vibrations and the soundness of the material. Otherwise, if resonance occured the usual results of fatigue and vibrations would follow. For this type of design

the classical and remarkable researches of *Campbell* (55) are particularly valuable, they have explained many questions concerning disc vibrations.

A large single-casing reaction turbine of typical *Westinghouse* type is shown in Fig. 146. It gives 65,000 kw. at 1800 R.P.M. with a single flow. It expands steam from about 300 lb./sq. in. (21 kg./cm.²) gauge and 700° F. (370° C.) down to a high vacuum. This machine and the previous one are probably the largest single-casing turbines ever built. However, a single-casing turbine of 80,000 kw. has recently been ordered and a turbine of as much as 100,000 kw. in one casing has been mentioned but does not yet appear to have passed the stage of design. The turbine shown is provided with a two-row velocity wheel for governing and a total of 22 reaction stages in groups, the last rows being mounted separately on curious disc shaped projections of a drum in two parts. The diameters increase up to 112 in. (2845 mm.) and the mean blade

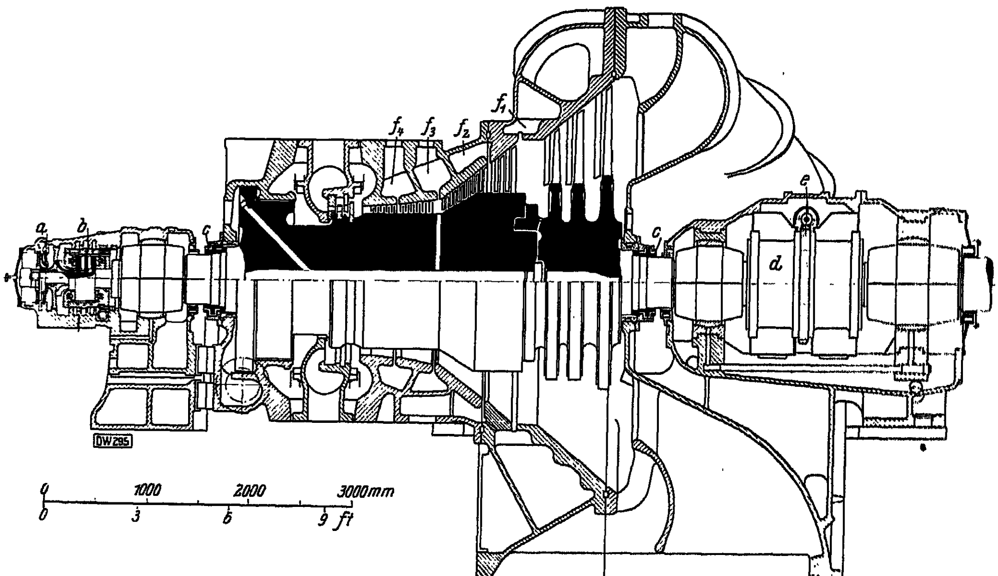


Fig. 146. *Westinghouse*, 65,000 kw., 1800 R.P.M. high-pressure turbine

a = Oil impeller for lubrication
b = Oil impeller for governing, combined with thrust bearing
c = Water-sealed gland
d = Double toothed type coupling
e = Barring gear
$f_{1 \div 4}$ = Bleeder branches for feed-heating

velocities will reach as much as 879 ft./sec. (267 m./sec.). At the tips of the blades the exceedingly high speed of nearly 1150 ft./sec. (350 m./sec.) is reached. The quality figure, calculated from the diameters and the steam conditions, is about 14,000 (2350). The only test results which have been published for a machine of this type include for four-stage feed-heating at pressures of about 171 (12), 74 (5.2), 15.7 (1.1) and 7.1 lb./sq. in. (0.5 kg./cm.²) absolute. The steam and heat consumptions are given, but it is only possible to estimate the efficiency. The value will be about 76% inclusive of all losses between the stop valve to the alternator terminals. Assuming a generator efficiency of 96.5%, about 79% would be the efficiency at the coupling.

B.B.C. build combined impulse-reaction turbines in one casing up to 50,000 kw. at 1500 R.P.M. of the type shown in Fig. 147. The turbine is almost an exact copy, to a larger scale, of the impulse-reaction turbines the firm used to build for 3000 R.P.M.. There is a similar arrangement of a number of impulse discs, seven in this case, followed by seven reaction discs. Overload steam is introduced before the third stage. The wheels are forced on to the

(55) Refer to foot-note 46 on p. 96.

shaft separately. The thrust is taken up by a small balance piston at the H. P. end and a thrust block in the governor end pedestal. The alternator coupling is of the claw type.

The latest principles in design of the *Brit. G.E.C.* were shown previously in an example of a double-casing turbine in Fig. 133. They are visible also in the

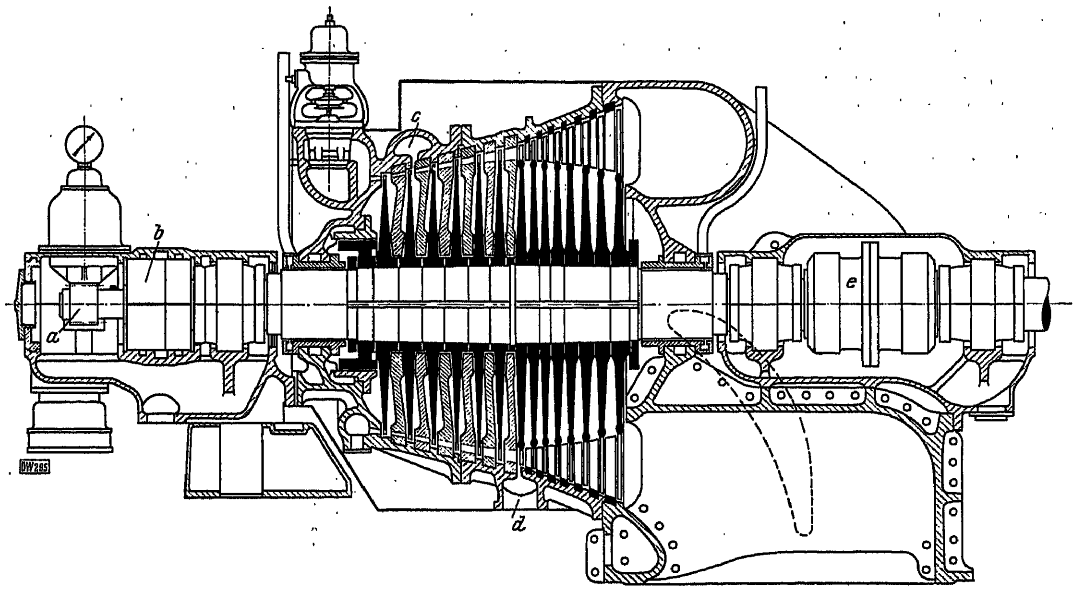


Fig. 147. *B.B.C.*, 30,000 to 50,000 kw., 1500 R.P.M. high-pressure turbine

a = Governor drive
b = Thrust block of the pad type
c = By-pass belt
d = Bleeder branch for feed-heating
e = Double claw-type coupling

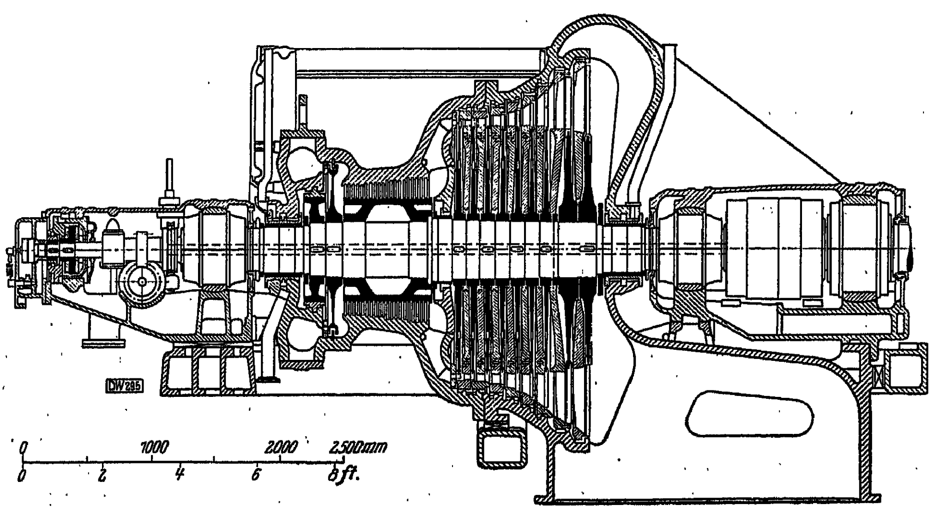


Fig. 148. *Brit. G.E.C.*, 18,750 kw., 1500 R.P.M. high-pressure turbine

18,750 kw. single-casing turbine in Fig. 148. After a two-row velocity wheel, there are 20 impulse stages of small diameter which, contrary to the usual practice, are carried on a drum. They are followed by nine impulse discs of much larger diameters. The turbine takes steam at 250 lb./sq. in. (17.5 kg./cm.^{2}) gauge, 650° F. (345° C.) and the vacuum is about 96.8%. The quality figure is about

9550 (1600). A steam consumption of 9.84 lb./kw.-h. (4.46 kg./kw.-h.) has been obtained by tests (56) at the economical load of 16,850 kw.. The corresponding efficiency is about 80% at the coupling. The machine has very small variations in efficiency at different loads. It may be noted in particular that the drum, the disc of the balance piston and the H.P. casing are in steel. In the L.P. part provision has been made for draining the water in the manner described on page 46. The *Brit. G.E.C.* have executed this design for machines up to 30,000 kw. and they recommend it for outputs up to 40,000 kw., arranged, however, for a better quality figure and for a higher efficiency.

It may be necessary to employ slow speed turbines for outputs below the limit value of high speed machines on account, for instance, of the alternator. If the best value of u/c_0 is required, relatively small diameters may have to be chosen in order to obtain sufficient blade heights. An increase in the number of stages will follow and it will often be necessary to build a double-casing machine. In this case, naturally, the steam flow will not be divided in the L.P. part.

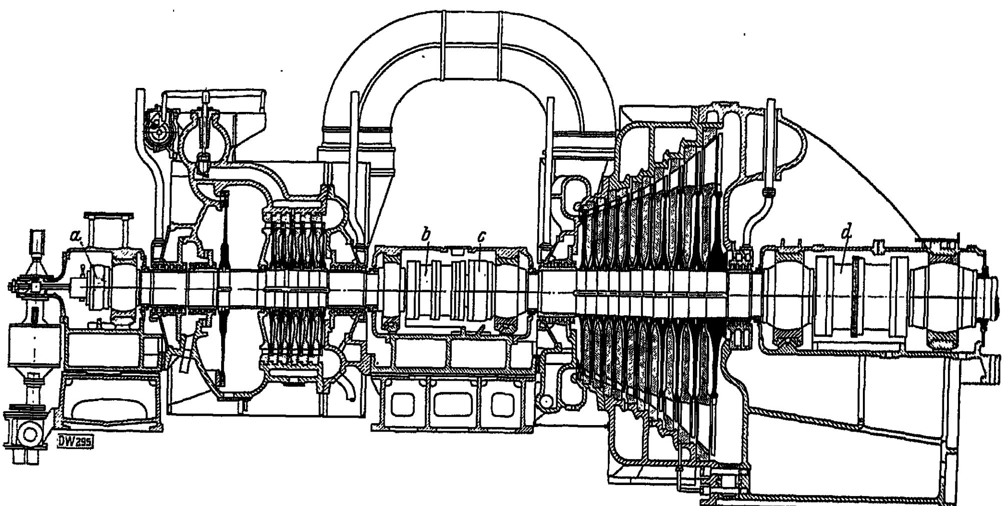


Fig. 149. *B.T.H.*, 32,000 kw., 1500 R.P.M., two-casing high-pressure turbine

a = H.P. thrust block of the multi-collar type
b = Double claw-type coupling
c = L.P. thrust block of the multi-collar type
d = Double claw-type coupling with gear wheel of barring gear

A double-casing *B.T.H.* turbine for single-flow may be seen in Fig. 149. According to the steam conditions it may be used for loads up to 32,000 kw. at 1500 R.P.M.. It has single-row impulse wheels throughout. The entire H.P. casing and the front part of the L.P. casing are in cast steel in spite of a moderate steam temperature. Both shafts run below the critical speed and are coupled together and to the alternator by means of claw-type couplings. Each shaft, therefore, has its own thrust block which may be adjusted for obtaining the correct axial clearance. Each journal bearing rests on the housing by the intermediary of packers inclined at 45°. This allows the shaft easily to be aligned. A 25,000 kw. turbine of this type gave an efficiency of nearly 81% at the terminals (57) or approximately 84% at the coupling. The steam conditions are about 214 lb./sq. in. (15 kg./cm.²) gauge, 665° F. (350° C.) and a moderate vacuum. It may also be mentioned that this turbine has as many as 12 throttle valves. This is contrary to the usual tendency of to-day which is to decrease the number of valves.

---

(56) See The Engineer 147 (1929) p. 202.
(57) See The Engineer 147 (1929) p. 284.

This machine may be compared with a *Metro-Vick* turbine for the same output (Fig. 150). It is designed for almost identical steam condition, the vacuum, however, is only about 92%. The H.P. part has a double-row velocity stage and 14 impulse wheels; the L.P. part consists of two groups of different diameters having eight and six discs respectively. The diameter of the last stage is 84 in. (2130 mm.), the Σu^2 is about 5,160,000 ft.^{2}/sec.^{2} (480,000 m.^{2}/sec.^{2}). The separate shafts are joined with flexible couplings, and it may be noted that the corrugated pipe type, as described on page 67 is used between the L.P. shaft and the alternator. The diaphragms for the H.P. casing and first eight L.P. stages are of mild steel plating with built up nozzles. All the moving blades are of stainless steel. A feed-heater is built into the bottom half of the exhaust casing, a well-known feature in *Metro-Vick* turbines. The *Baumann* multiple exhaust has not been employed.

Intermediate types between single-flow and double-flow machines in two casings exist for large slow speed turbines as well as for 3000 R.P.M. sets. They

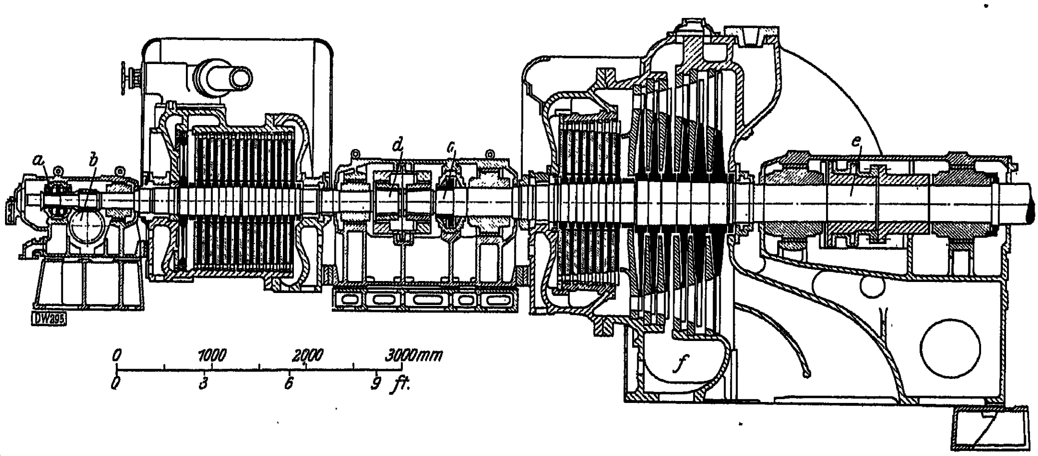


Fig. 150. *Metro-Vick*, 25,000 kw., 1500 R.P.M., two-casing high-pressure turbine
a = H.P. thrust block of the pad type   c = L.P. thrust block of the pad type   e = Coupling of the corrugated pipe type
b = Governor drive   d = Double claw-type coupling

may be made in a single casing with a double-flow L.P. end, the last stages being duplicated, or they may be of the *Baumann* multiple exhaust type.

The method employed by the *Brit. G.E.C.* in the single-casing turbine of Fig. 54 *d* is also used by *Allis-Chalmers* for obtaining a double flow in turbines for limit capacities (Fig. 151). The example shown is for 60,000 kw. and is designed for pure reaction. The L.P. part has two groups of stages which work in parallel. They are placed one behind the other and exhaust into the same opening. Instead of the reaction stages being arranged on a drum, the same method is used as in the latest *B.B.C.* and *Parsons* machines, and they are mounted in groups or separately on discs. These are forced on to the rotor. Throttling is, naturally, the only method of governing a pure reaction turbine. The machine works in conjunction with a primary turbine for 1200 lb./sq. in. (84 kg./cm.^{2}) steam pressure. Reheating is arranged between the two sets and four branches are provided for feed-heating.

In Fig. 152 may be seen an example of a 1500 R.P.M. *Metro-Vick* double-casing turbine for 44,000 kw.. It is very similar to the high-speed turbine in Fig. 131. A flexible coupling of the corrugated pipe type, a feed-heater built into the exhaust casing and all the characteristic details in designs of the firm may be seen.

Naturally, there is greater scope for increasing the size of two-casing turbines with double-flow L.P. end. If there are no special conditions the factor limiting the size of single-shaft units is the maximum capacity of the generator. The only differences between these machines and 3000 R.P.M. double-casing double-flow machines are the larger dimensions.

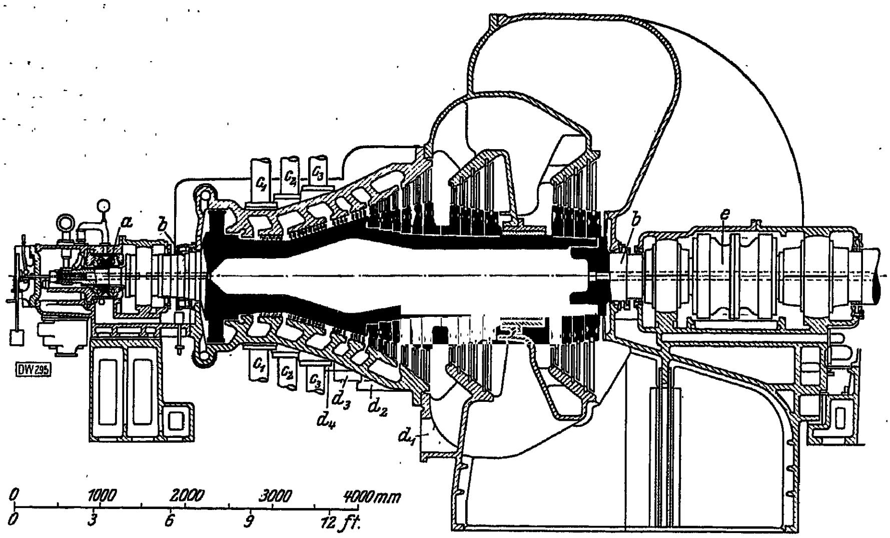


Fig. 151. *Allis-Chalmers*, 60,000 kw., 1800 R.P.M., single-casing, double-flow high-pressure turbine

a = Thrust block of the pad type
b = Water-sealed gland
$c_{1 \div 3}$ = Live steam pipes
$d_{1 \div 4}$ = Bleeder branches for feed-heating
e = Double claw-type coupling

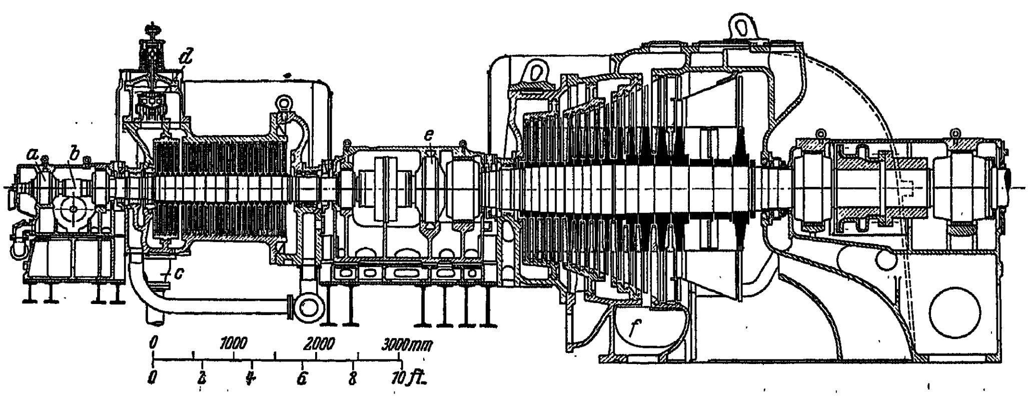


Fig. 152. *Metro-Vick*, 44,000 kw., 1500 R.P.M., two-casing high-pressure turbine with multiple exhaust

a = H.P. thrust block of the pad type
b = Governor drive
c = Live steam pipe
d = Overload valve
e = L.P. thrust block of the pad type
f = Feed-heater

In Fig. 153 may be seen the largest plain impulse turbine of this type. It is, probably, the largest single-shaft turbine ever built. It is of the *Amer. G.E.C.* and gives 160,000 kw. at 1500 R.P.M.. The H.P. part has a two-row velocity wheel followed by 17 single-row impulse stages. The L.P. part has two groups of six discs. The last stage has a mean diameter of about 123 in. (3125 mm.). The working conditions are about 375 lb./sq. in. (26.5 kg./cm.²) gauge, 700° F. (370° C.) and 0.5 lb./sq. in. (0.035 kg./cm.²) absolute back-pressure.

Fig. 154 shows the largest turbine built by *Allis-Chalmers*. Like all machines of this firm, it is for pure reaction. With about 600 lb./sq. in. (42 kg./cm.²) gauge, 725° F. (385° C.) and a good vacuum, 65,000 kw. may be obtained at 1800 R.P.M.. Both shafts are bored through and are joined with a double claw-type coupling. All the steam is extracted after the second group of H.P. stages and is sent to a reheater. There are several branches for feed-heating. At the

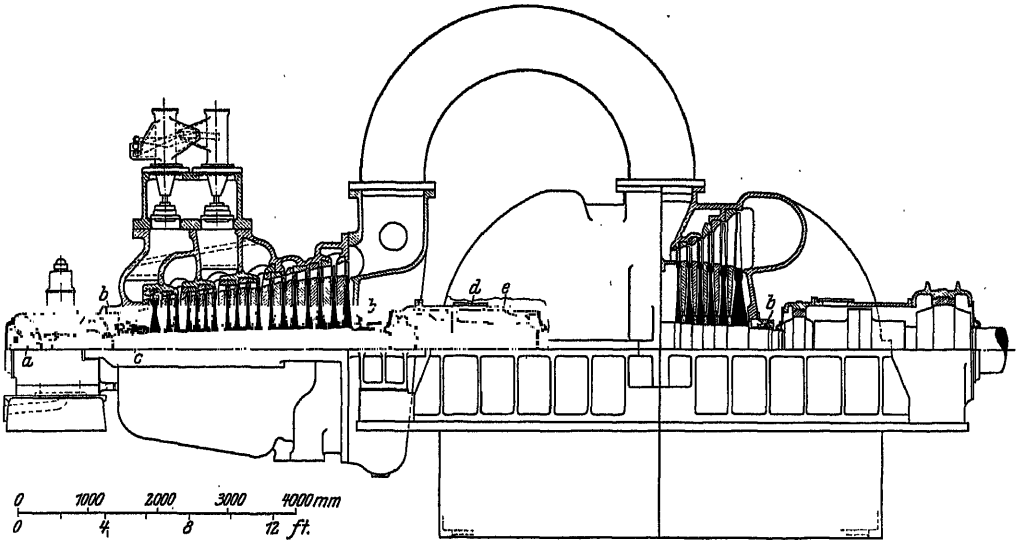


Fig. 153. *Amer. G.E.C.*, 160,000 kw., 1500 R.P.M., two-casing high-pressure turbine

a = H.P. thrust block of the multi-collar type
b = Water-sealed gland
c = Three-flow labyrinth-type packing gland
d = Claw-type coupling
e = L.P. thrust block of the multi-collar type

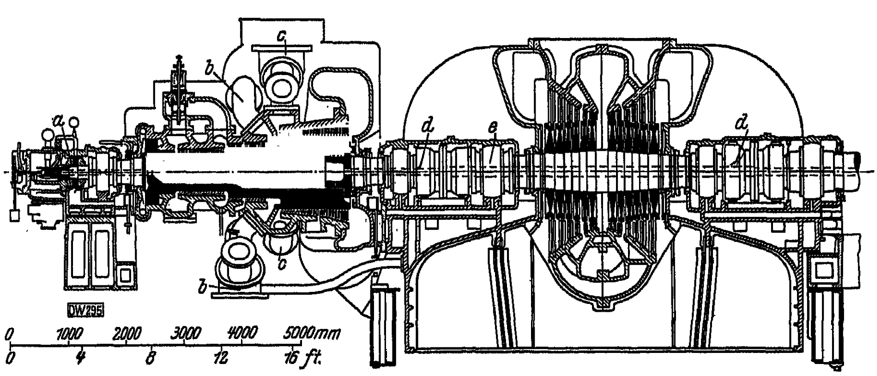


Fig. 154. *Allis-Chalmers*, 65,000 kw., 1800 R.P.M., two-casing, double-flow high-pressure turbine

a = H.P. thrust block of the pad type
b = Extraction branch for reheating
c = Inlet of steam from reheater
d = Double claw-type coupling
e = Combined journal and thrust bearings

present time *Allis-Chalmers* have a 115,000 kw. turbine of similar design under construction. It is for the same steam conditions and will be the largest single-shaft unit for 1800 R.P.M..

The 85,000 kw. three-casing *B.B.C.* turbine in Fig. 155 greatly resembles the machine in Fig. 143, which is also a three-casing turbine, but for 3000 R.P.M.. It is designed for only 192 lb./sq. in. (13.5 kg./cm.²) gauge and 680° F. (360° C.) and with its large number of stages it will have a very good quality figure. The

L.P. stages have a diameter of 114 in. (2900 mm.). With the exception of the first stage, the turbine is of the reaction type.

In Fig. 156 may be seen an *A.E.G.* turbine for the same power station and of the same capacity. The steam conditions are identical. The turbine is in

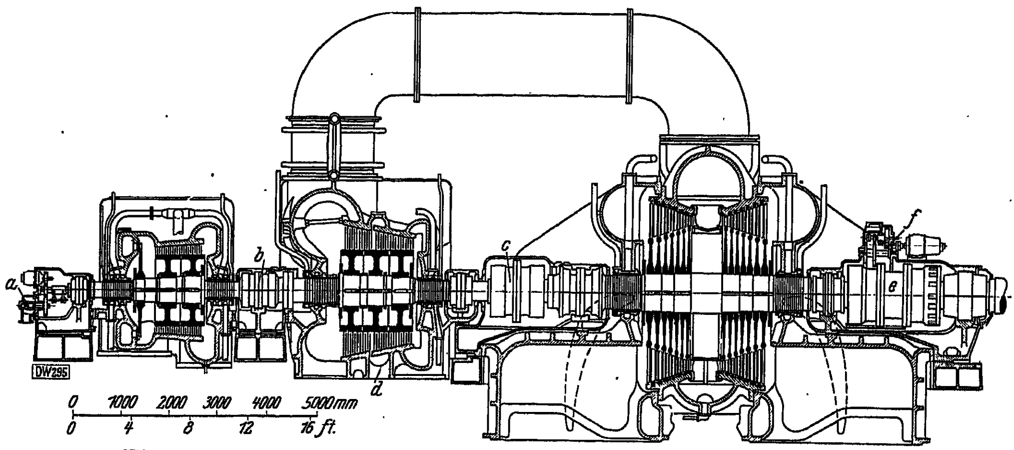


Fig. 155. *B.B.C.*, 85,000 kw., 1500 R.P.M., three-casing, double-flow high-pressure turbine

a = Oil pump
b = Journal bearing combined with double thrust block of the pad type
c = Double claw-type coupling
d = Bleeder branch for feed-heating
e = Double claw-type coupling
f = Barring gear

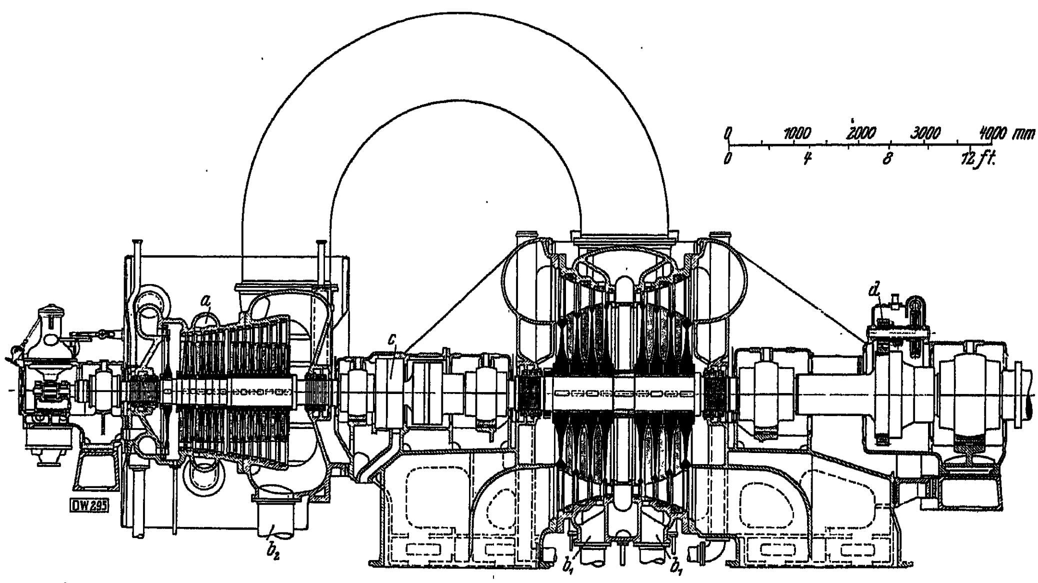


Fig. 156. *A.E.G.*, 85,000 kw., 1500 R.P.M., two-casing high-pressure turbine

a = By-pass belt
$b_{1,2}$ = Bleeder branches for feed-heating
c = Combined H.P. and L.P. thrust block of the pad type
d = Barring gear

two casings and all the stages have discs. In the H.P. part there are 14 single-row impulse wheels of about 71 in. (1800 mm.) diameter and in the L.P. part two sets of four reaction stages of 114 in. (2900 mm.) diameter.

In order to be completely independent of any technical limit to the size of a turbine set it is necessary to abandon the tandem design and adopt a cross-

compound arrangement. It will then be possible to have as many casings and generators as desired. Outputs requiring this method are so high that in Europe it has only been necessary to employ it twice, at the Klingenberg and at the Barking Super Power Stations. The first of three identical sets at Klingenberg was put into operation at the end of 1926. The turbines are for 80,000 kw. and were built for the best efficiency obtainable at the time. The working conditions are 462 lb./sq. in. (32.5 kg./cm.^{2}) gauge, 750° F. (400° C.) and 96% vacuum. The *A.E.G.* adopted a twin shaft design with two 44,000 kv.-a. alternators each driven by two turbines. The H.P. set has H.P. and I.P. turbines and the L.P. set has two L.P. turbines working in parallel (refer to sketch on left of Fig. 57). Machines for the same output but with a single shaft and two or three casings, have already been described. The advantages of a four-casing unit with two shafts are the reduced overall length, the greater accessibility, also the use of two alternators of only half the total capacity. No difficulties were encountered when starting up with two separate shafts, the electric coupling of the alternators allowing the turbine to be run up to speed as easily as any other machine of the same capacity. The $\Sigma\, u^2$ is 8,950,000 ft.^{2}/sec.^{2} (830,000 m.^{2}/sec.^{2}) and the quality figure about 17,900 (3000). The acceptance tests were carried out by *Josse* and proved, under the specified conditions, higher efficiencies than guaranteed (58). The first stage is a two-row velocity wheel. The H.P. and I.P. turbines are of the disc type and have impulse blading; the L.P. rotors have reaction drums. The steam flows through the H.P. and I.P. turbines and through the two L.P. turbines in opposite directions. This enables the thrust to be almost eliminated in the H.P. set, whilst the L.P. set is completely balanced. With regard to the governing, the first point to be settled was the method to be adopted: nozzle or throttle governing. A machine of this size is usually run at as high a load as possible and plain throttle governing might be considered the most appropriate. For such large quantities of steam, however, several valves have to be provided in any case and throttle governing looses its chief advantage, its simplicity. It was decided, therefore, to adopt nozzle governing. This gave also a 2% improvement in steam consumption at half load; the first stage being a two-row velocity wheel, was able to take good advantage of the larger heat drop obtained at partial loads. Thus, two stop valves were provided, followed by three throttle valves for full load and two for overload.

The demand for still larger units exists only in America. Three examples of recent turbines for enormous capacities will be described here. The two first are for 160,000 kw. and were built for the Hell Gate Power Station in New York. The third is the largest set ever made, the 208,000 kw. turbine for the State Line Generating Station. The first two machines were put into service early in 1929. They are of particular interest as a design produced in Europe, by *B.B.C.*, is in competition with one produced in America, by *Westinghouse*, and a most useful comparison may be made.

Both machines are two-cylinder cross-compound and have double-flow L.P. ends. They are reaction turbines, the first one is entirely designed for reaction, the second one has a velocity wheel for governing. The *B.B.C.* set has its H.P. part for 1800 R.P.M. and its L.P. for 1200 R.P.M.. The economical loading is between 55,000 and 90,000 kw.. The maximum continuous load is 160,000 kw.. Fig. 157 gives a section through the H.P. cylinder. The stages are in three groups of seven, eight and nine respectively; the diameters increase from 58.7 to 72.5 in. (1490 to 1840 mm.). Five discs are forced on to the shaft from the H.P. end; four are for carrying the blading, the fifth is the balance piston. A throttle valve admits steam before the first stage, enabling 55,000 kw. to be obtained. A second valve allows steam to enter the turbine in front of the

(58) See *W. E. Wellmann* Z. VdI. 72 (1928) p. 1077.

eighth stage and the output rises to 90,000 kw.. For obtaining 160,000 kw. two additional valves are provided. One of these is connected to the first admission belt and may be seen in the illustration on the top of the H.P. casing; the other is joined to the piping between the second throttle valve and the belt before the eighth stage. Both valves by-pass steam to the third group of stages. All the throttle valves are double-seated; the stop valve is single-seated. The complete system for governing has two groups of valves in parallel, the arrangement described above being in duplicate. The L.P. part is shown in Fig. 158. It has two sets of eight stages on seven discs; the mean diameters are from 135 to 155 in. (3440 to 3930 mm.). The last blade is 37.8 in. (960 mm.) long. At normal load about 52 tons per hour of water condense in the L.P.

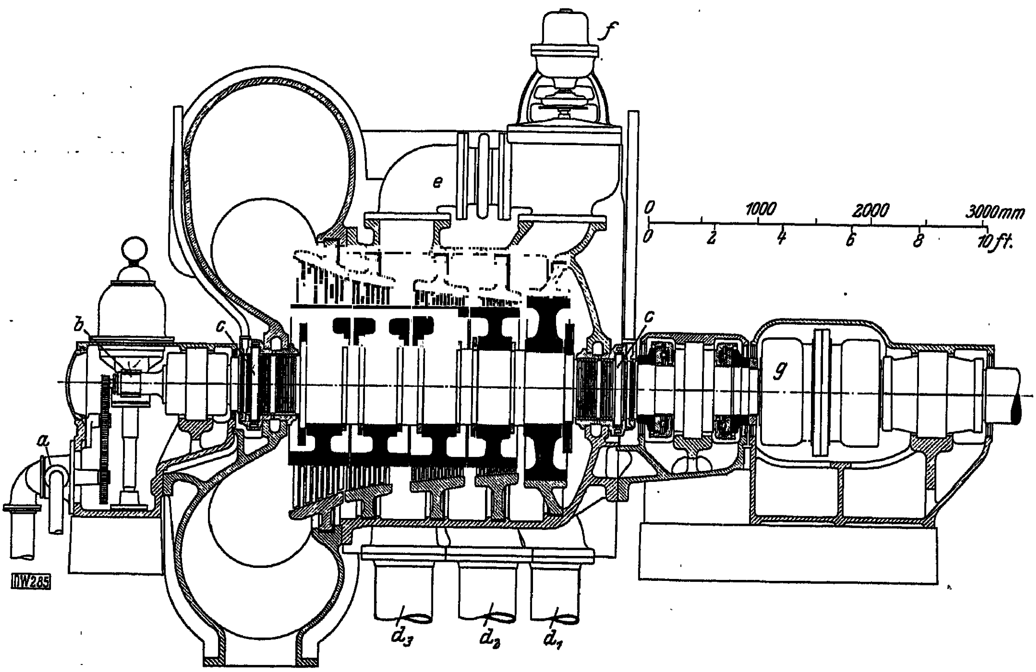


Fig. 157. *B.B.C.,* 1800 R.P.M. H.P. turbine of the 160,000 kw. set

a = Oil pump
b = Governor Drive
c = Water-sealed gland
$d_{1\div3}$ = Live steam connections
e = By-pass
f = Overload valve
g = Double claw-type coupling

end if 13% wetness is assumed. For draining this large quantity of water grooves are provided on the fixed blades after the second L.P. stage. They lead to holes communicating with annular spaces between the roots of the fixed blades and the casing. The water separated before the fourth L.P. stage is drawn off through a bleeder branch and at lower stages it is led to the exhaust opening. Feed-heating is provided after the H.P. turbine at 17.8 lb./sq. in. (1.25 kg./cm.^{2}) absolute and from the L.P. turbine at 3.6 lb./sq. in. (0.25 kg./cm.^{2}) absolute. The working conditions are 265 lb./sq. in. (18.6 kg./cm.^{2}) gauge, about 600° F. (320° C.) and 96.7% vacuum. The best guaranteed efficiency is 82.7% at the coupling and this value should be obtained at all loads between 55,000 and 90,000 kw.. The quality figure is 20,900 (3500) at 90,000 kw..

The *Westinghouse* set was ordered a short time later. It is shown in Figs. 159 to 161 and runs at one speed only, 1800 R.P.M.. The H.P. turbine drives an 80,000 kw. alternator and two 200 kw. exciters, the L.P. turbine

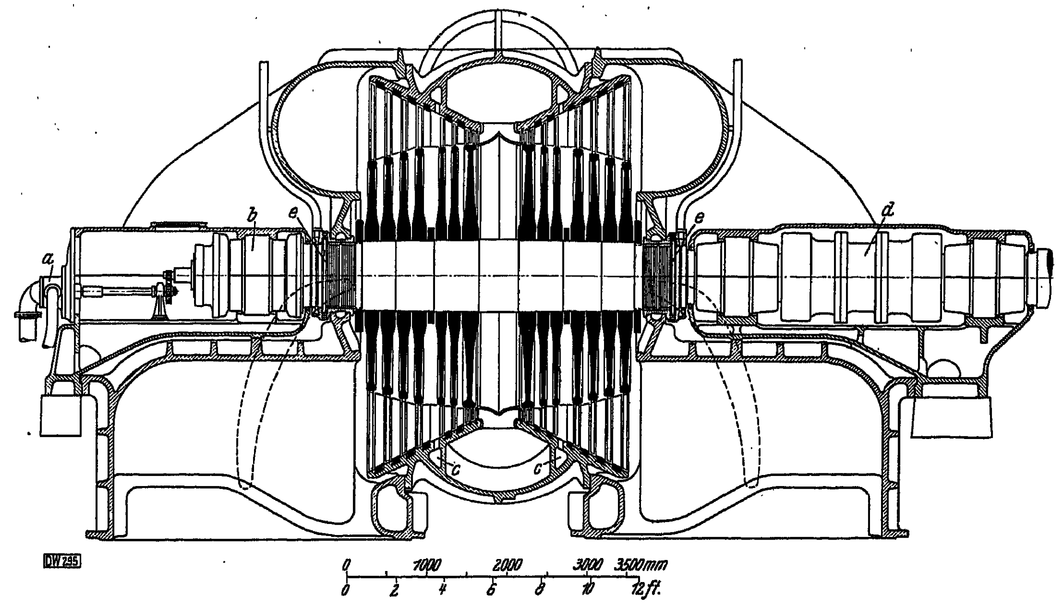


Fig. 158. *B.B.C.*, 1200 R.P.M. L.P. turbine of the 160,000 kw. set

a = Oil pump
b = Journal bearing combined with double thrust block of the pad type
c = Belt for bleeding feed-heating steam
d = Double claw-type coupling
e = Water-sealed gland

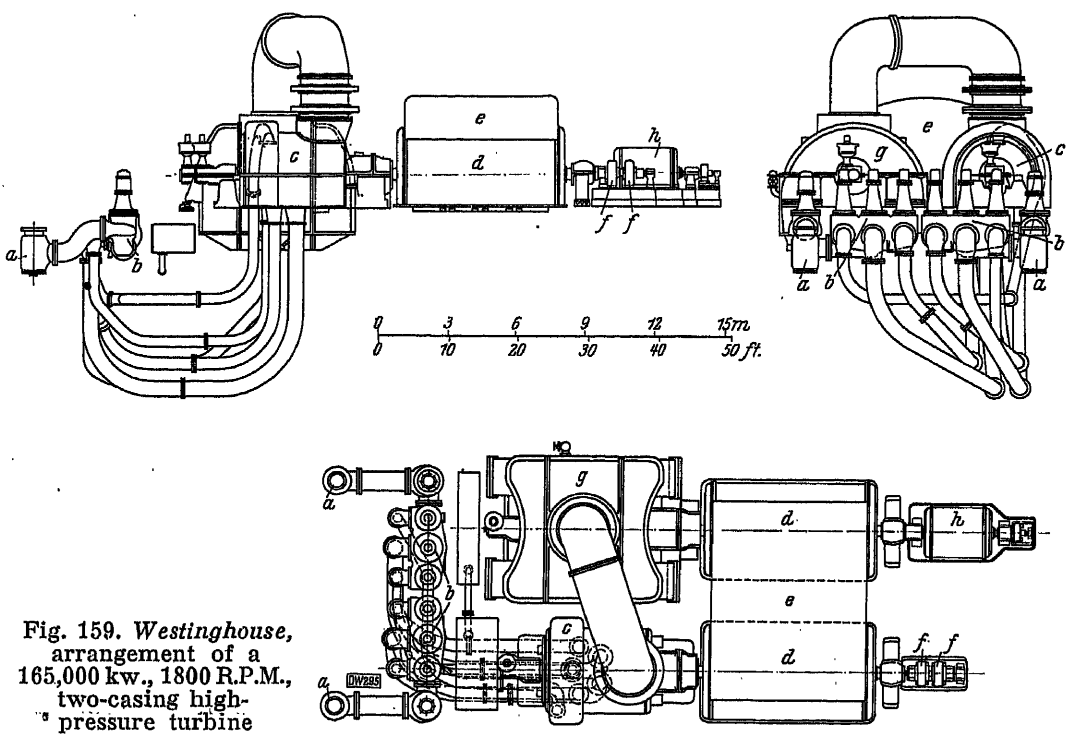


Fig. 159. *Westinghouse,* arrangement of a 165,000 kw., 1800 R.P.M., two-casing high-pressure turbine

a = Stop valve with steam strainer
b = Throttle valves
c = H.P. casing
d = Alternator, 80 000 kw.
e = Common shell of the two Alternators d
f = Exciter, 200 kw.
g = L.P. casing
h = House-set, 5000 kw.

drives another 80,000 kw. alternator, a 5000 kw. house set and an auxiliary exciter. The steam arrives from the boilers through two pipe lines with separate stop valves each supplying a group of three throttle valves. These are joined to the turbine by wide bends. The pressure and temperature being only moderate this arrangement shows the great care which was taken to avoid any

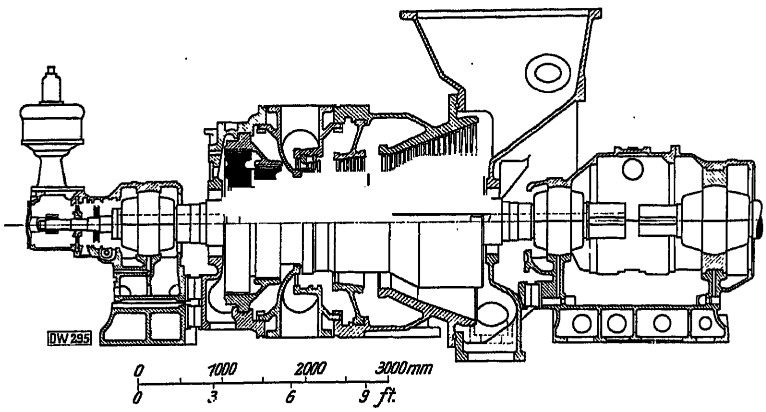


Fig. 160. *Westinghouse*, H.P. casing of the 165,000 kw., 1800 R.P.M. two-casing turbine

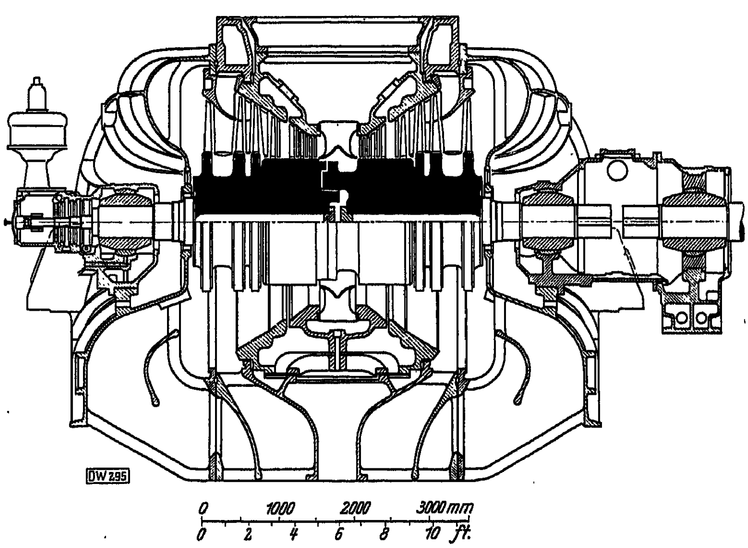


Fig. 161. *Westinghouse*, L.P. casing of the 165,000 kw., 1800 R.P.M. two-casing turbine

harmful thrust from the piping on the turbine. Except for 90° F. (50° C.) higher temperature, the steam conditions are the same as for the other set.

The H.P. turbine (Fig. 160) has a two-row velocity wheel and 14 reaction stages in two groups. The diameters increase from 51 to 76 in. (1300 to 1935 mm.). Although the centre distance is about 16 ft. (5 m.) the rotor is a solid drum. It is bored along the axis. Two balance pistons of different diameters and a *Kingsbury* thrust bearing are provided for taking up the thrust.

The L.P. turbine in Fig. 161 has the characteristic *Westinghouse* multiple exhaust for the last stages. Two sets of seven stages are mounted on a drum in two parts. The last three rows are on discs cut from the solid. The mean diameters increase up to 97 in. (2460 mm.). The steam flow is divided in the stage before the last, only part of the steam expanding in the last stage. It is only possible to allude here to other peculiar details of construction such as the exhaust casing with its numerous ribs inside and out, and its carefully designed diffusor shape, the side entry blades of the last three stages (refer to Fig. 42 *k*) and many other novel features. The machine is also arranged for two-stage feed-heating. The quality figure is about 15,000 (2500), which is considerably less than the *B.B.C.* set. Unfortunately, no guarantees or test results have been published yet and it is not possible to give a comparison of the steam consumptions.

The largest turbine ever built is for a maximum load of 208,000 kw., the economical loading being between 105,000 and 150,000 kw.. It was put into

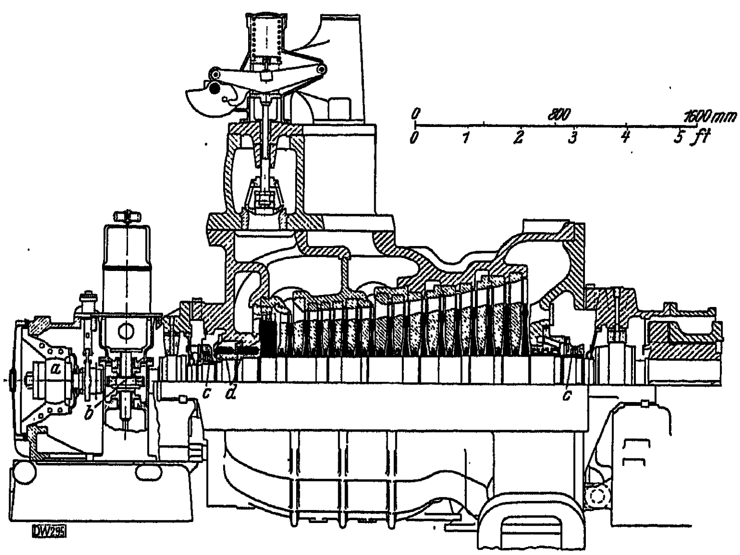


Fig. 162. *Amer. G.E.C.*, H.P. casing of the 208,000 kw., 1800 R.P.M. three-casing set

a = Thrust block of the multi-collar type
b = Governor and oil pump drive
c = Water-sealed gland
d = Three-flow labyrinth-type packing gland

service at the end of 1929 and was built by the *Amer. G.E.C.* for the new State Line Generating Station. It has two expansions and three casings and shafts (refer to Fig. 57, right). The H.P. part is for a maximum load of 76,000 kw.. Parallel to it and on either side are two double-flow L.P. turbines of 66,000 kw. each. All the shafts rotate at 1800 R.P.M.. The H.P. turbine drives an alternator and exciter. The main alternators of the L.P. turbines are for 62,000 kw. only, they are directly coupled, however, to 4000 kw. house sets. The steam conditions are 600 lb./sq. in. (42 kg./cm.^{2}) gauge, 730° F. (390° C.) and the vacuum is 97%. After the H.P. turbine all the steam is sent to a reheater and is raised to a temperature of 500° F. (260° C.) before entering the L.P. turbines. Five bleeder branches have been provided for feed-heating. The set should have a heat consumption of 9750 B.Th.U./kw.-h. (2460 kcal./kw.-h) exclusive of the losses in the boilers, but inclusive of reheating and feed-heating. The calculation was made with a turbine

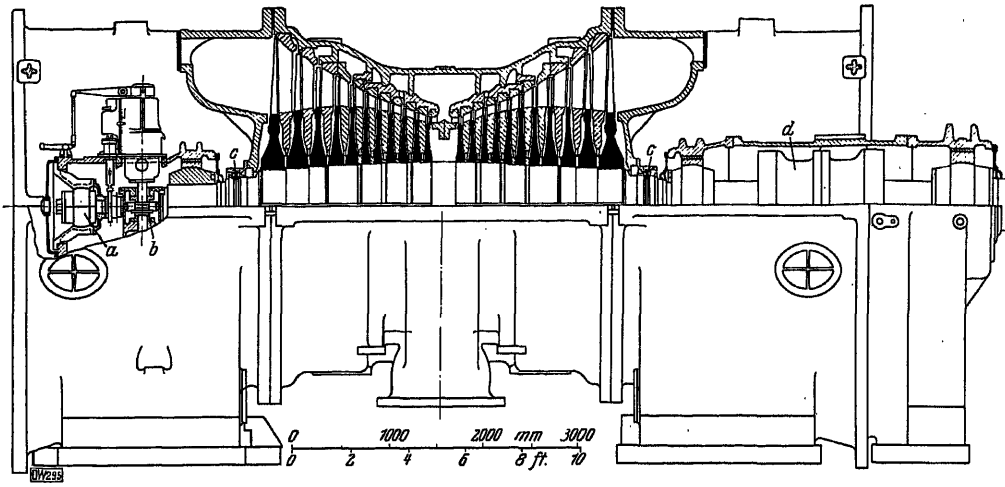


Fig. 163. *Amer. G.E.C.*, L.P. casing of the 208,000 kw., 1800 R.P.M. three-casing set

a = Thrust block of the multi-collar type
b = Governor and oil pump drive
c = Water-sealed gland
d = Double claw-type coupling

efficiency corresponding to the average quality figure of 13,750 (2300). The set would be less economical at partial loads and overload.

With the exception of the last L.P. stages the blading is designed for plain impulse. The H.P. part (Fig. 162) has a velocity stage and 16 single-row impulse wheels. The mean diameters rise to about 63 in. (1600 mm.) and the length of the last blade is 11 in. (280 mm.). The pressure after the first stage is high and a labyrinth gland in three sections has been provided. The two L.P. turbines are identical (Fig 163); the steam expands from the middle towards either end of the turbine and passes through 11 single-row impulse stages. These are designed for small amounts of reaction, which increase towards the exhaust. The mean diameters rise to 104 in. (2650 mm.) and the blade lengths to 30 in. (760 mm.). Fig. 164 shows the H.P. turbine and the throttle valves. The extremely thick flanges at the horizontal joint may also be seen. The set has an unusual appearance as each L.P. turbine has four vertical condensers; the size of these and of the exhaust branches almost make the turbine look insignificant.

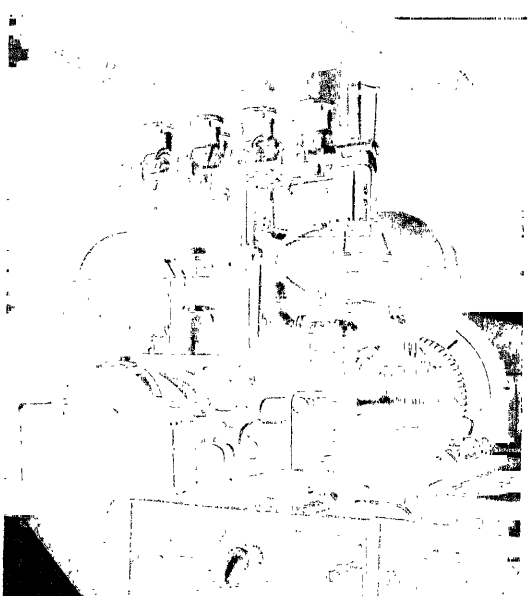


Fig. 164. *Amer. G.E.C.*, H.P. casing and governing arrangement of the 208,000 kw., three-casing turbine set in the shops

Vertical condensers reduce the price of the foundations. They greatly increase the height of the engine room, however,

and completely alter the arrangement of the L.P. casing. These points are apparent in Fig. 165 which gives a horizontal section through a *Westinghouse* L.P. turbine with condensers of this type. It is probable that a vertical condenser will have a somewhat lower efficiency than one of the horizontal type. The condensate will trickle down the tubes and will increase the resistance to the heat flow from the steam to the circulating water. There will also be considerable undercooling as the condensate is so long in contact with the tubes, and, especially, the lower tube sheet which is cooled by the coolest water. Before giving a decided opinion, however, it is necessary to wait until accurate practical data are obtained. A detail of design which may be mentioned is that the flanges between the turbine casing and the condenser are inclined. This facilitates the removal of the top part of the turbine casing.

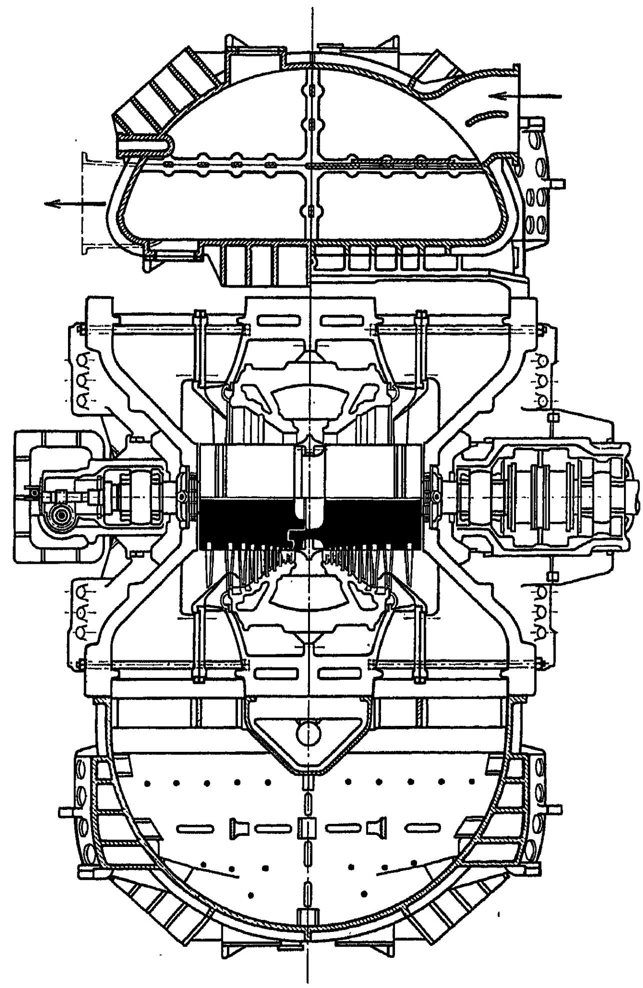


Fig. 165. *Westinghouse,* view of the horizontal joint of a double-flow L.P. turbine with vertical condensers. Output 20,000 kw.; speed 1800 R.P.M.

b. Back-pressure turbines

Only part of the energy contained in the steam is extracted in back-pressure turbines, the remainder is used in the form of heat for warming, drying or other purposes. The turbines are placed, therefore, between the boilers and the process. The latter usually requires a constant pressure. The steam exhausting from turbines being free from oil, applications for back-pressure turbines are continually becoming more numerous.

If the initial and exhaust pressures are fixed, the load and the steam quantity are in direct dependence. Thus, if the steam quantity is given, the load is determined. It will not be possible, therefore, to govern these two quantities independently. The governing must depend either on the pressure in the process main or on the speed, it cannot depend on both.

If the demand for process steam is to determine the governing the pressure after the turbine will be regulated. When more steam is required the pressure

in the process main will fall and the constant back-pressure governor will open the turbine throttle valve. On the other hand, the pressure will rise when less steam is required and the valve will close. The load will vary according to the steam flow and the speed must be maintained by coupling the generator to a power grid. If the steam quantity falls too low to enable the required amount of power to be obtained, the deficiency must be made good by taking current from the grid. The more frequently this occurs or the larger the quantity of outside current required, the greater will be the importance of the price paid for power. If, on the other hand, an excess of power is produced from the process steam, it may, naturally, be sold cheaply.

A different method of operation would be for a factory, having a large demand of power, to sell the surplus steam at a cheap rate. Public power and heating stations, for instance, are plants of this kind. In this case the demand for steam, however, varies greatly with the season. There are at present but few large stations for distributing steam to industrial concerns for drying, cooking or other uses having only small seasonal fluctuations.

When the load is governed, the rise in the demand for power will reduce the speed, this affects the governor, the throttle valve opens and the steam quantity and output increase. When the load falls the reverse operations take place. This method of governing is satisfactory, therefore, if more steam is needed than is necessary for producing the power required. If the additional demand for steam is small, if it is only of short duration or if it occurs only infrequently, the deficiency can be made good by opening a reducing valve between the high-pressure and the process mains. When large quantities of steam are required separate L.P. boilers should be provided for supplying the deficiency of steam directly to the process.

If the average demand for steam and power are balanced, temporary variations occurring however, a steam accumulator will be economical. The simplest arrangement will allow excess steam to be stored with only a slight rise in pressure and steam may be released at falling pressures. Many methods of connecting accumulators have been devised and patented but they will not be discussed here as they do not belong to the subject of the present book (59).

When designing a back-pressure turbine the starting point is the volume of steam. Naturally, other factors of importance are the speed which is chosen, the output obtained and the number of stages and casings. If the volume of steam is very small, as will occur with high initial and final pressures and small steam quantities, the blade lengths may be so small, the steam jets so narrow and the leakage and surface friction losses so great that it will not be possible to build a multi-stage turbine of sufficiently good efficiency. This point is illustrated in Fig. 21. It will be advisable in such cases to use a single wheel of small diameter, running at a high speed and having partial admission. As is well known, single-stage machines used to be the only form of back-pressure turbine. They possess the quality of having only small variations in efficiency with different steam quantities or outputs. Hence, it will still be correct to use a back-pressure turbine with a single impulse or velocity stage when large fluctuations occur in the steam flow. As an application, reference may be made to the modern single-stage back-pressure turbines for great speeds, small loads and high pressures. A turbine of this kind, for a small volume of steam, will be described later (refer to Fig. 199).

The steam flow may be approximately constant and the average volume may be large enough to allow a multi-stage turbine, running at a correctly chosen speed, to have blades of sufficient length with full admission. It will then be

---

(59) Refer to *Th. Stein:* "Regelung und Ausgleich in Dampfanlagen", especially chapter VII (Berlin: J. Springer 1926).

possible to build multi-stage turbines for small outputs and good internal efficiencies. Nevertheless, all multi-stage turbines are considerably less efficient when the steam quantity or load is reduced, the no-load consumption being especially high.

When the number of stages is increased the maximum efficiency of a back-pressure turbine usually rises, the drop in efficiency at partial loads, on the other hand, will be greater. The rise in efficiency through an increase in the number of stages will be less for turbines taking small volumes of steam at full load. When the volume is very small it may happen that the maximum efficiency obtained with a large number of stages is less than with a single stage. These principles may be seen from the curves given in Fig. 166. The lower curve shows the internal efficiency (inclusive of wheel friction losses but exclusive of losses in the bearings) of a back-pressure turbine with only a two-row velocity wheel. The middle curve is for a turbine with 14 stages and a quality figure of about 8370 (1400); the turbine for the top curve has 21 stages and about 11,060 (1850) quality figure. The fall in efficiency at partial loads will be greater, therefore, for turbines with higher quality figures.

These questions are of fundamental importance for all multi-stage turbines. It is common practice to maintain a constant back-pressure; the specific volume of the exhaust steam will then remain unchanged, or perhaps vary slightly. When it is desired to use less steam than the designed amount in a multi-stage back-pressure turbine, the total volume of the exhaust steam will decrease and its velocity in the last stage will diminish. The heat drop and the output of this stage will be only small. If, for example, the steam quantity is reduced to half, the total volume and the velocity will also be halved, but the power obtained in the last stage from a unit weight of steam will be only about a quarter. As is well known, the stage pressure in the first stages of the turbine rises according to an elliptical law and the heat drops, the steam velocities and the stage efficiencies will increase gradually. The final result is that the total efficiency of multi-stage back-pressure turbines is low at partial loads, a fact which has just been seen from curves. The objection may be avoided by designing the blading for a smaller load than full, such as ¾ load. The fall in efficiency then will only occur from below this load or steam quantity and will be smaller. Full load will be obtained in the same way as an overload, the highest efficiency will be at ¾ load and the steam consumptions at partial loads and no-load are improved. An important conclusion is arrived at concerning the design; high efficiency back-pressure turbines should never be too amply rated, it is preferable to design them for a too small than for a too high load. In the case of small multi-stage turbines this rule cannot be applied always as too short blades might be obtained. For this

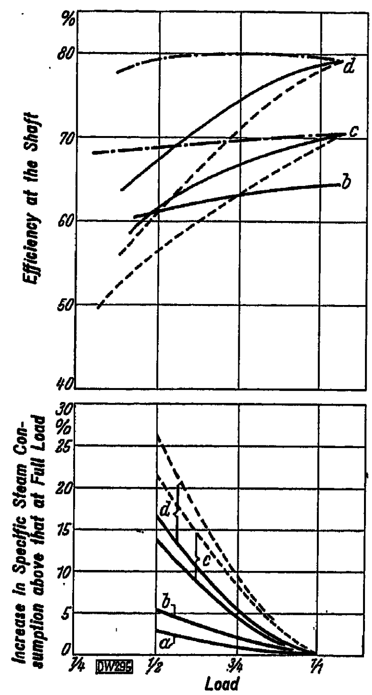


Fig. 166. Efficiency and steam consumption of back-pressure impulse turbines of different quality figures, $q = \frac{\Sigma u^2}{h_o}$, at various loads

a = Two-row velocity wheel, $c_o/u = 5$
b = Two-row velocity wheel, $c_o/u = 4$
c = Multi-stage impulse turbine $q = 8370 \frac{\text{ft.}^2/\text{sec.}^2}{\text{B.Th.U./lb.}}$ $\left(1400 \frac{\text{m.}^2/\text{sec.}^2}{\text{kcal./kg.}}\right)$
d = Multi-stage impulse turbine $q = 11{,}600 \frac{\text{ft.}^2/\text{sec.}^2}{\text{B.Th.U./lb.}}$ $\left(1850 \frac{\text{m.}^2/\text{sec.}^2}{\text{kcal./kg.}}\right)$
—— Nozzle governing } referred to the
–·–· Throttle governing } total heat drop
—·— Throttle governing, referred to the throttled heat drop

reason it is more important to know the exact operating conditions of a multi-stage than of a single-stage turbine. If the steam consumption is known only approximately, either a single-stage machine should be chosen or the design should be arranged to allow an easy reconstruction if the conditions are different from those expected. Small multi-stage back-pressure turbines are for indirect drive and will be described later (Figs. 206 and 207).

Fig. 166 shows in the case of the three machines the superiority of nozzle over throttle governing for back-pressure turbines (the curves are compiled from test results) (60). With throttle governing the internal efficiency referred to the heat drop in the turbine varies only slightly at partial loads, even for multi-stage machines, and it may possibly start by rising, as in the top curve, and only begin to fall sharply after half load. If, however, the efficiency is referred to the initial heat drop it will fall very rapidly and decrease approximately as the ratio of the heat drop in the turbine to the initial heat drop. The efficiency of the same machine with nozzle governing will also fall with the load and steam

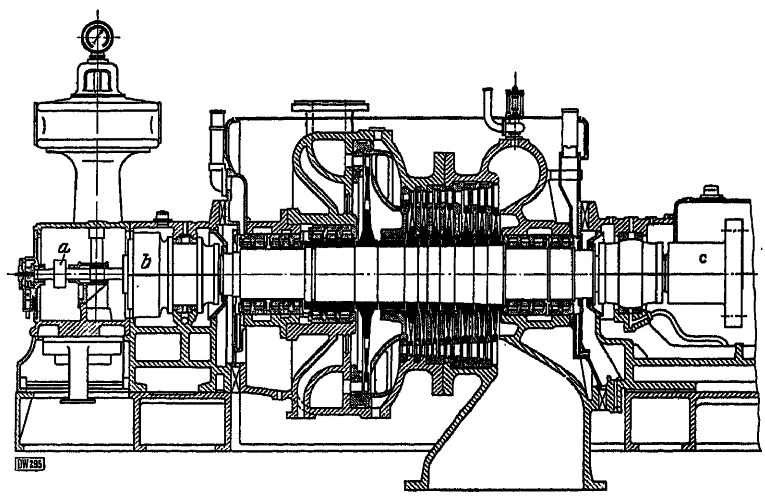


Fig. 167. *Bergmann,* 7500 kw., 3000 R.P.M. back-pressure turbine
a = Emergency governor b = Two-collar thrust bearing with pivoted pads c = Flanged coupling (half)

quantity, but it will do so less rapidly. These relations may be seen more clearly from the lower group of curves in the illustration. The percentage increase in steam consumption at partial loads above the value at full load is shown for the three machines. The velocity wheel will have at ¾ load an increase of only 1 to 2% according to the design, the turbine with 8370 (1400) quality figure has 4.5% for nozzle governing and 8.5% for throttle governing, whilst the turbine with 11,060 (1850) quality figure has 5.2 and 9.5% increases. Consequently, throttle governing will be applicable to back-pressure turbines only in special cases. There would be no object, for instance, in designing a turbine for a low steam consumption and increasing the price if the power extracted from the process steam could not be completely utilized and steam from the boilers had to be passed into the process main directly through a reducing valve.

A few examples of back-pressure turbines will now be briefly described. They will show that the method of expansion is of no consequence, provided the design is correct. Apart from one exception, all the examples are pro-

(60) Concerning questions on governing of lesser importance at the present day for condensing turbines refer to *H. Baer:* "Die Regelung von Dampfturbinen und ihr Einfluß auf die Leistungsentwicklung in den einzelnen Druckstufen". Forschungsarbeiten No. 86 (Berlin: VDI.-Verlag 1910).

vided with a single or a double-row impulse wheel for regulation, throttle governing not being suited for back-pressure turbines.

Fig. 167 gives a 7500 kw. *Bergmann* back-pressure turbine with seven impulse stages. The characteristic details employed by the firm are plainly visible in the illustration and require no comment.

The *A.E.G.* turbine in Fig. 168 is for somewhat lower outputs and for higher pressures. It is provided with a two-row velocity wheel. It expands steam from 455 lb./sq. in. (32 kg./cm.²) gauge, 750° F. (400° C.) down to 256 lb./sq. in. (18 kg./cm.²) gauge and gives 2000 kw. at 3000 R.P.M.. The high back-pressure classifies it as a „primary" turbine. Machines of this type may be used when the boiler pressure is raised in a large power station; they are placed between the new high-pressure boilers and the old steam main.

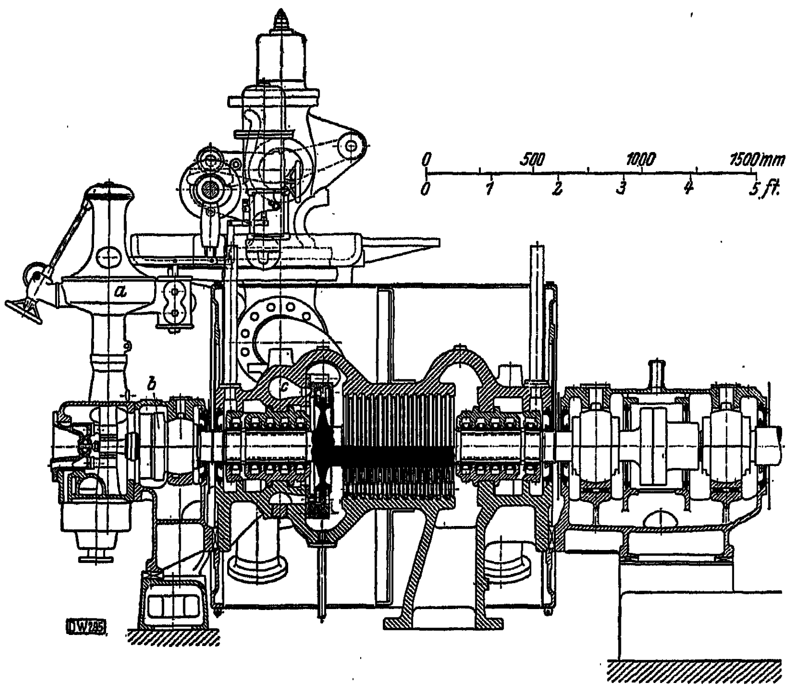


Fig. 168. *A.E.G.*, 2000 kw., 3000 R.P.M. back-pressure turbine
a = Speed governor b = Thrust block of the pad type c = Steam belt

An eleven-stage back-pressure turbine of the *de Laval* impulse type is shown in Fig. 169. It gives 4000 kw. at 3000 R.P.M. and is for steam at 780 lb./sq. in. (55 kg./cm.²) gauge.

When the heat drop is large and the steam quantity not subject to wide fluctuations, a two-casing back-pressure turbine may be considered. A machine of this kind is shown in Fig. 170. It is another plain impulse turbine and is for 7500 kw. at 3000 R.P.M.. It is designed for 850 lb./sq. in. (60 kg./cm.²) gauge, 840° F. (450° C.) and 71 lb./sq. in. (5 kg./cm.²) gauge exhaust pressure. Two machines of this type of only 3200 kw. capacity for 256 lb./sq. in. (18 kg./cm.²) gauge, 660° F. (350° C.) and 10 lb./sq. in. (0.7 kg./cm.²) gauge back-pressure gave a full load efficiency of 84% at the coupling at the acceptance tests in a chemical works in Central Germany.

A few reaction turbines will now be mentioned and they may be compared with the impulse turbines which have just been described. A small *B.B.C.* reaction turbine will also be illustrated later when dealing with turbines for indirect drive (page 160).

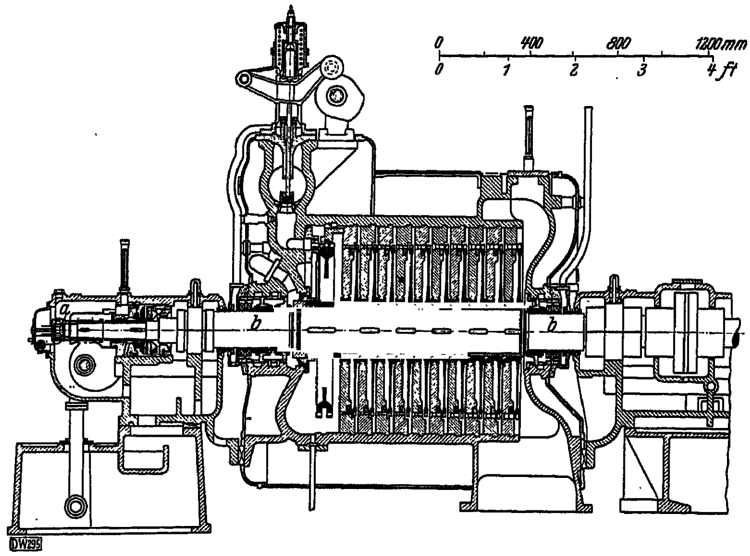


Fig. 169. *De Laval,* 4000 kw., 3000 R.P.M. back-pressure turbine
a = Emergency governor   b = Combined labyrinth and carbon-type packing gland

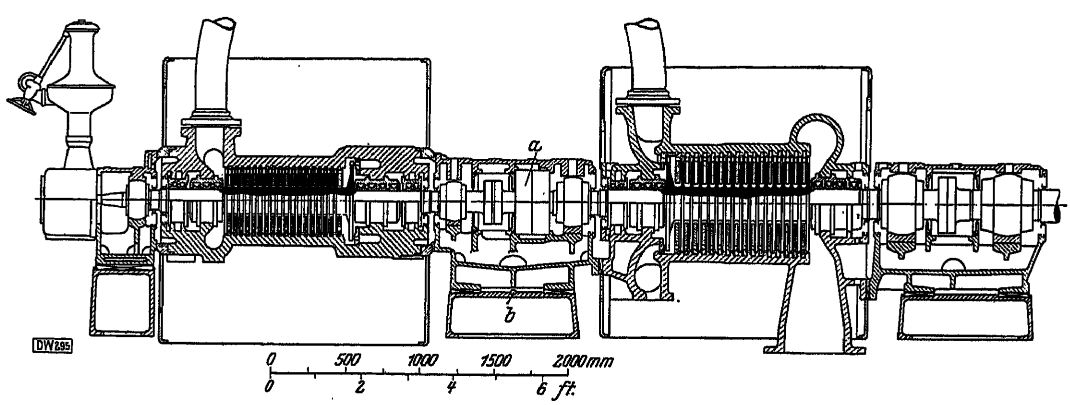


Fig. 170. *A.E.G.,* 7500 kw., 3000 R.P.M., two-casing back-pressure turbine
a = Thrust block of the pad type for the H.P. and L.P. rotors   b = Fixed point

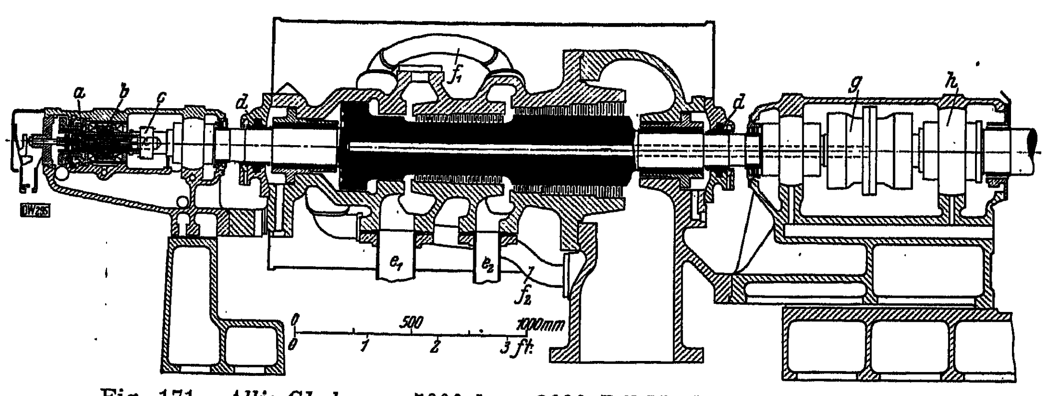


Fig. 171. *Allis-Chalmers,* 5000 kw., 3600 R.P.M. back-pressure turbine

a = Emergency governor
b = Thrust bearing of the pad type
c = Governor drive
d = Water-sealed gland
$e_{1,2}$ = Live steam pipes
$f_{1,2}$ = Steam pipe to balance piston
g = Double claw-type coupling
h = Journal bearing with double thrust block

A pure reaction back-pressure turbine is shown in Fig. 171. The first stage has full admission and below full load the governing is by throttling. A second valve, on the same spindle as the first, admits overload steam before the second group of stages. *Allis-Chalmers* build turbines of this type up to 5000 kw. at 3600 R.P.M., a lower speed being adopted for greater outputs. The firm has already made machines for initial pressures up to 540 lb./sq. in. (38 kg./cm.^{2}) gauge, 700° F. (370° C.) and about 125 lb./sq. in. (8,8 kg./cm.^{2}) gauge back-pressure.

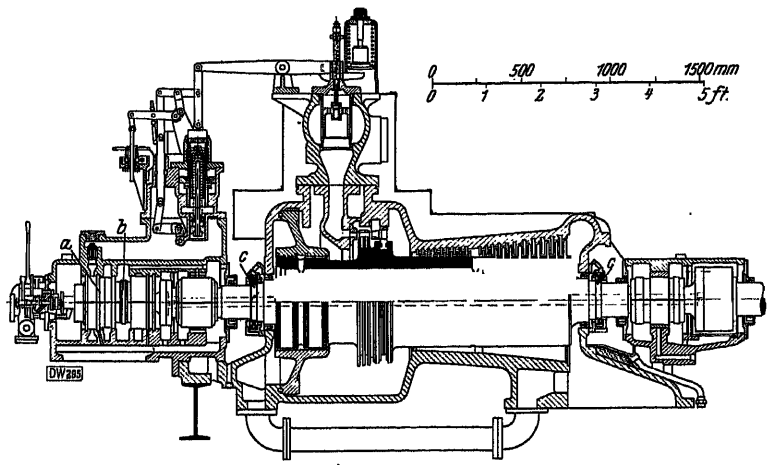


Fig. 172. *Westinghouse,* 4000 kw., 3600 R.P.M. back-pressure turbine
a = Oil impeller for governing   b = Thrust bearing of the pad type   c = Water-sealed gland

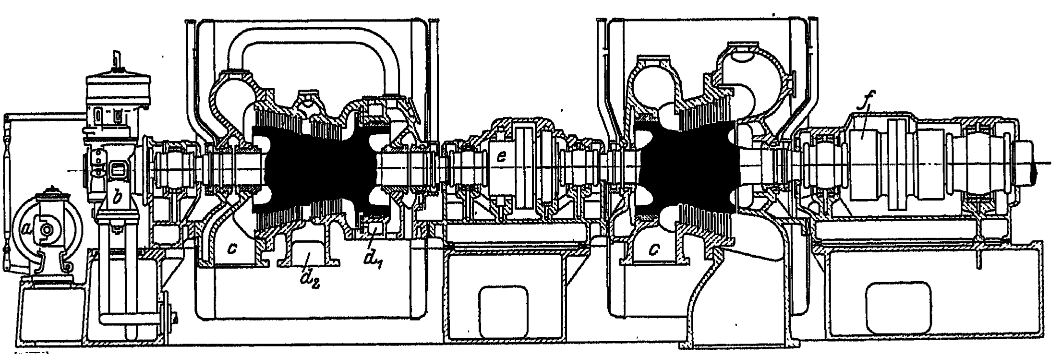


Fig. 173. *S.S.W.,* 16,000 kw., 3000 R.P.M., two-casing back-pressure turbine

a = Servo-motor of throttle valve
b = Oil pump
c = Branches for the pipe connecting the H.P. and L.P. casings
$d_{1,2}$ = Live steam pipes
e = Flanged coupling with thrust bearing of the pad type
f = Double claw-type coupling

A larger reaction back-pressure turbine is shown in Fig. 172. It was built by *Westinghouse* for 400 lb./sq. in. (28 kg./cm.^{2}) gauge, 660° F. (350° C.) and 21 lb./sq. in. (1.5 kg./cm.^{2}) absolute back-pressure and gives 4000 kw. at 3600 R.P.M.. The 20 stages are mounted on a solid drum. All the details of construction are in accordance with the usual *Westinghouse* methods.

As a last example of a back-pressure turbine, a machine will be mentioned which may be compared to the design in Fig. 170. It is a two-casing *S.S.W.-Roeder* turbine for 16,000 kw. at 3000 R.P.M. (Fig. 173). The flow is in different directions in the two cylinders in order to balance the thrust. In front of the H.P. drum is a single-row impulse wheel for governing.

c. Extraction turbines

Extraction turbines are condensing or back-pressure machines from which process steam may be drawn. There may be a single extraction branch or several; in the second case steam will be obtained at different pressures.

The simplest form of extraction turbine has no pressure governing. The pressure will then vary in proportion to the flow of steam through the part of the turbine following the extraction branch; it will depend, therefore, on the load and quantity of steam extracted. This type of turbine is not of great importance as an industrial machine, it has been extensively employed of late, however, with one or several branches for feed-heating. The steam bled in this case is only a relatively small amount and the machines do not differ greatly in design from ordinary high-pressure or back-pressure turbines.

Extraction turbines with pressure governing at the extraction branch have a H.P. or back-pressure part followed by a L.P. part. The steam not extracted is expanded down to a vacuum or a given back-pressure in the L.P. part, which is regulated by a separate governor. The same principles apply whether the H.P. part is single or multi-stage, whether it is in one casing with a partition between the two parts or in two casings, a type which is now often used. The load, the heat drop and the efficiency required, in the H.P. part especially, are the main factors determining the design. The governing of the H.P. part, which is that of the total steam quantity, and of the flow through the L.P. part are performed by speed and pressure governors.

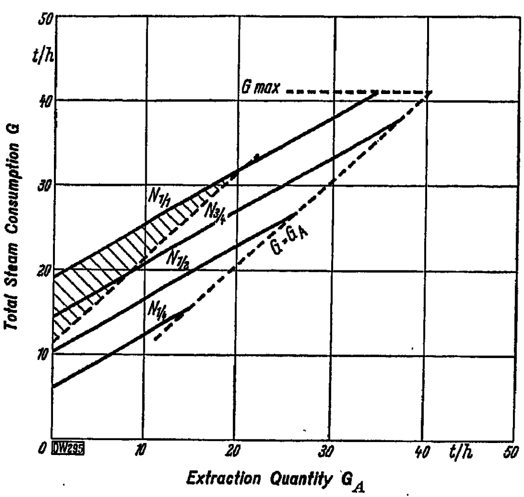


Fig. 174. Steam consumption curves of an extraction turbine for 3200 kw. at 3000 R.P.M.

Rise in extraction pressure

The relations between the total steam consumption, the extraction quantity and the load are usually given by means of diagrams (Fig. 174). The limits are fixed by the design of the machine. The maximum total steam consumption, $G_{max.}$, is the greatest amount of live steam the turbine has been designed to pass, whilst the largest amount of steam which may go to the condenser is the quantity for which the L.P. part has been calculated. The diagram shows also that even within these limits certain conditions of operation may only be obtained under particular circumstances. In the first place, the turbine will work as a back-pressure machine when the extraction quantity becomes equal to the total steam consumption. The line limiting the curves on the right represents this mode of operation. Since the turbine is working as an ordinary back-pressure machine, any increase in extraction quantity will necessarily raise the output and an independent governing of the load is no longer possible. Vacuum should be maintained in the L.P. part of the turbine in order to reduce the ventilation losses. To the left of the back-pressure line the turbine works as a genuine extraction machine and when the extraction pressure is kept constant the load and extraction quantity may be governed independently. The L.P. part will deal with the steam not required for the process but necessary for producing a sufficient output. The lower limit of this region is given by the dotted line on the left, which separates a small shaded portion of the diagram. If the load is

great and the extraction quantity small, obviously a large amount of steam must flow to the condenser. Therefore, the pressure before the L.P. part will rise and will exceed finally the specified value for the process. Conditions of operation corresponding to any of the points in the shaded region all imply a rise in extraction pressure. If a constant pressure is required, either the extraction quantity must be increased, any excess in steam which cannot be used in the process being perhaps blown off into the atmosphere, or the load must be decreased, the deficiency being made good from another source.

Hence, in the case of all extraction turbines, it is of the greatest importance to have a thorough knowledge of the operating conditions. Once the process and live steam pressures have been chosen, the following points have to be settled: (1) the maximum extraction quantity at full load, (2) the minimum extraction quantity at full load, (3) the maximum load required when no process steam is being used, the turbine working as an ordinary high-pressure or back-pressure machine.

If the load to be obtained without extraction has been chosen too high the L.P. part will be very large. For ordinary conditions of operation it will be uneconomical as it will contain very little steam and the pressure before its first stage will be small. On the other hand, if the extraction quantity has been chosen too high, the H.P. part will be too large and will only work with a good efficiency if it is single stage. The more stages it has, and the higher its efficiency at full load, the more uneconomical will it be for the usual conditions. Apart from very large units, extraction turbines to meet greatly varying demands of process steam should only have few H.P. stages. It is most important also only to demand guarantees for load and extraction quantities which will frequently occur during operation. If the designer has to give many widely different points, he will have to adopt a less satisfactory design and a worse efficiency for normal operation in order to meet the exceptional requirements. Great attention should be paid to these points as it unfortunately happens very often that extraction turbines are ordered for conditions different from those occurring in practice.

Fig. 175 shows a standard *A.E.G.* extraction turbine for moderate outputs and steam conditions. When a total of about 62,000 lb./h. (28,000 kg./h.) is flowing through the H.P. part and 33,000 lb./h. (15,000 kg./h.) is being extracted at 21.3 lb./sq. in. (1.5 kg./cm.^{2}) gauge the turbine gives 3600 kw. at 3000 R.P.M.. The H.P. part has a two-row velocity wheel and the L.P. a single-row impulse wheel for regulation. Both these stages have partial admission. The remainder of the stages consist of a moderate number of impulse wheels which are cut from the solid in the H.P. part and are forced on to the shaft in the L.P. part owing to their larger diameters. The two regions are separated by a diaphragm with several throttle valves.

Figs. 176 and 177 show similar machines of different manufacture, but differing only slightly from the one just described. The first example is a single-casing extraction impulse turbine of the *English Electric*. The extraction valves are placed at the side of the machine and are not visible on the sectional arrangement. Fig. 177 shows an application of the methods of standardization of the *Amer. G.E.C.* which have already been mentioned (p. 101). The illustration gives a double-extraction machine which is formed by assembling separate segments, the last one, for instance, being exactly the same as for the turbine in Fig. 121. Another original feature is the method of governing, a circular slide valve being used instead of the ordinary valves. The nozzles of the stages following the extraction branches can be shut off or diminished in section according to requirements. The method has been known for a long time and it shows again that in America moderate sized turbines, such as all industrial turbines, are regarded somewhat as if they were only small machines from which only medium efficiencies would be expected. This point of view is certainly correct in so far as a trifling fault in the heating system, with its

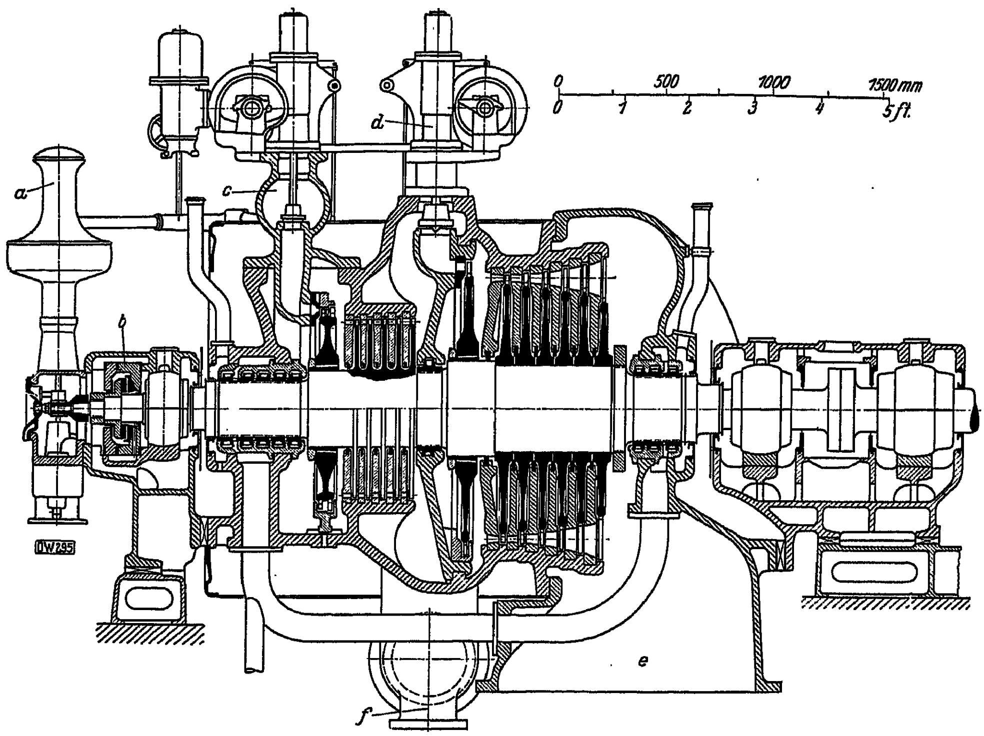


Fig. 175. *A.E.G.,* 3600 kw., 3000 R.P.M. extraction turbine

a = Speed governor
b = Thrust block of the pad type
c = Live steam regulating valve
d = Extraction regulating valve
e = Exhaust branch
f = Extraction branch

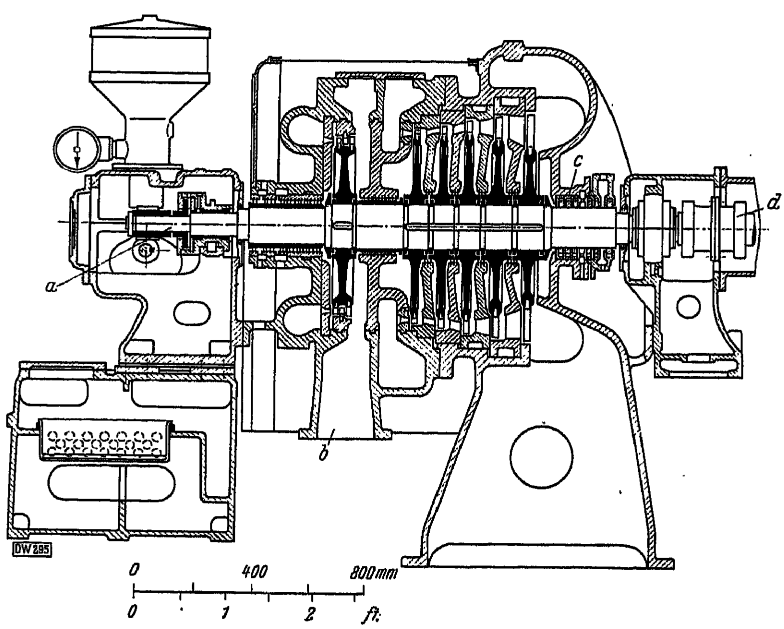


Fig. 176. *English Electric,* 1375 kw., geared extraction turbine for 6000/3000 R.P.M.

a = Thrust block of the pad type
b = Extraction branch
c = L.P. packing gland of the carbon type
d = Double toothed type coupling

numerous heaters and complicated arrangement, may result in a much greater loss than a slightly less turbine efficiency.

Allis-Chalmers also use pure reaction turbines with throttle governing for extraction machines (Fig. 178). Types have already been developed for outputs up to 10,000 kw. at 3600 R.P.M.. In this case also, full load steam enters before the first group of stages and overload steam before the second group.

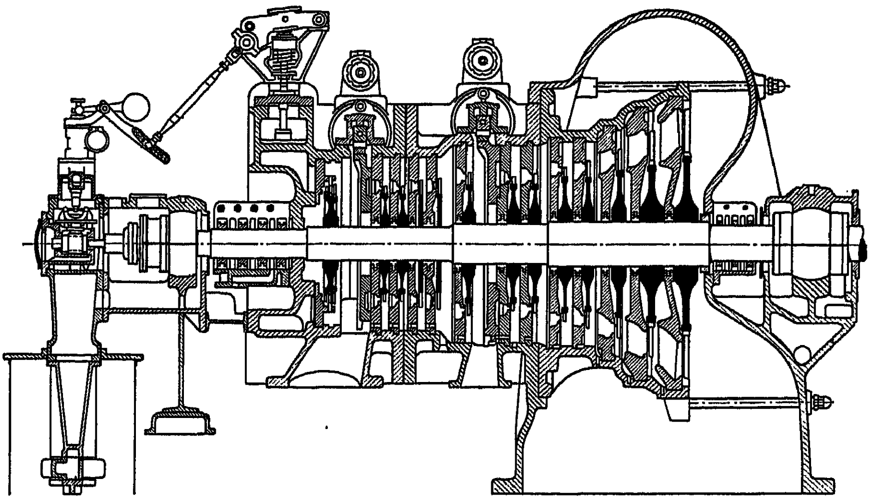


Fig. 177. *Amer. G.E.C.,* 3600 R.P.M. extraction turbine with two governed extraction points

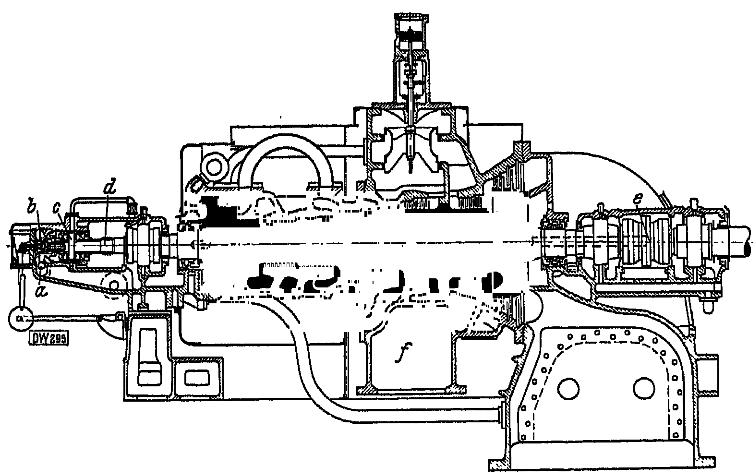


Fig. 178. *Allis-Chalmers,* 10,000 kw., 3600 R.P.M. extraction turbine

a = Thrust block adjusting gear
b = Centrifugal emergency governor
c = Thrust block
d = Worm wheel driving governor
e = Double claw-type coupling
f = Extraction branch

For larger heat drops and fairly constant demands for process steam, extraction turbines are also built in two casings. A good place for the extraction branch is after the first casing, or earlier if the extraction pressure is high. An example of this type of machine is given in Fig. 179 which shows a 2500 kw. *Erste Brünner* turbine. The H.P. part has 19 single-row impulse stages on a solid rotor, the L.P. part has a drum with 30 reaction stages preceeded by a single-row impulse stage for regulation. All throttle valves are placed at the side of the turbine. The live steam is at 300 lb./sq. in. (21 kg./cm.²) gauge, 750° F. (400° C.).

When the different steam quantities of the two turbine parts are considered it may often be found an advantage to choose different speeds for the separate turbines of a two-casing extraction machine. As will be seen later, the arrangement may easily be realized by means of gears.

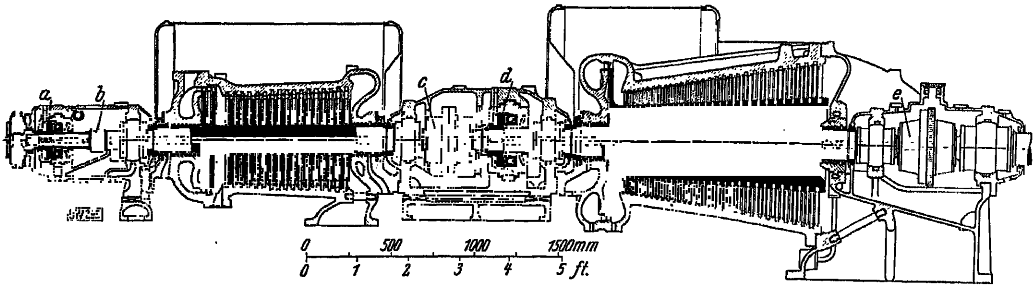


Fig. 179. *Erste Brünner*, 2500 kw., 3000 R.P.M., two-casing extraction turbine

a = H.P. thrust block with pads resting on balls
b = Governor drive
c = Double claw-type coupling
d = L.P. thrust block with pads resting on balls
e = Double claw-type coupling with gear wheel of barring gear

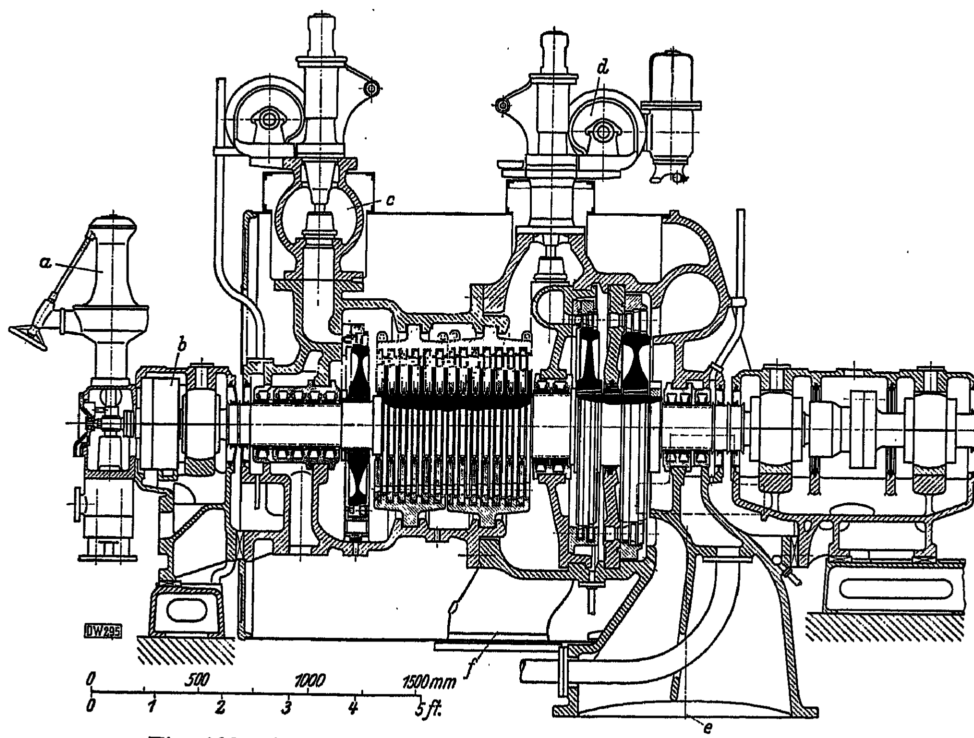


Fig. 180. *A.E.G.*, 3500 kw., 3000 R.P.M. extraction turbine

a = Speed governor
b = Thrust bearing of the pad type
c = Live steam regulating valve
d = Extraction regulating valve
e = Exhaust branch
f = Extraction branch

When the average extraction quantity is large most of the output is produced in the H.P. part, and the L.P. part may be designed for less steam than if the load were to be obtained with no extraction. Usually, the generator is only fully loaded when the extraction is large, this also will enable the size of the L.P. part to be reduced. The fundamental principle is that the turbine should always be designed for the best efficiency at the most frequent conditions of operation, it being, of course, necessary to consider as well the questions of

maximum load *and* extraction, also that of maximum load with *no* extraction. The turbine may then have a design as in Fig. 180 in special cases. This machine is in a spinning mill and takes steam at 325 lb./sq. in. (23 kg./cm.^{2}) gauge, 700° F. (375° C.). The extraction pressure is 28 lb./sq. in. (2 kg./cm.^{2}) gauge. The H.P. part was designed of the multi-stage type as the extraction quantity is always large, being about 83,000 lb./h. (37,500 kg./h.) at the rated load. The L.P. part, on the other hand, has two double-row velocity wheels only as the turbine runs but occasionally purely condensing and then no more than half load is wanted.

When process steam at two different pressures is required and a condensing part is unnecessary, extraction back-pressure turbines may be used. As in the case of ordinary back-pressure turbines, two methods of governing are possible. When the set is connected electrically to other machines, it will not matter if the output varies and it will be possible to maintain the pressures constant in the two process mains: the extraction and the exhaust. If this is not possible and the extraction back-pressure turbine is the only generating unit in the system, the pressure may be governed in *one* main only, whilst in the other it will vary according to the demand for process steam and the load. The steam consumption will depend on the extraction quantity and on the output, the latter being controlled by the speed governor. The governing will work then as in an extraction condensing turbine.

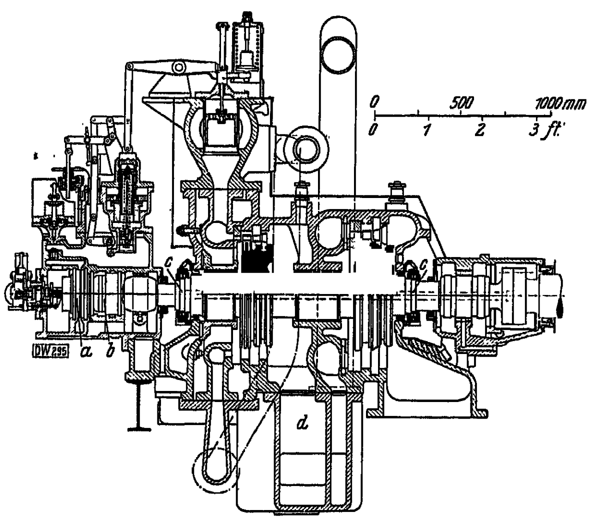


Fig. 181. *Westinghouse,* 7500 kw., 3600 R.P.M., extraction back-pressure turbine

a = Oil impeller for governing c = Water-sealed gland
b = Thrust bearing of the pad type d = Bleeder branch for feed-heating

The extraction back-pressure turbine in Fig. 181 is well suited for highly fluctuating conditions of operation. It gives 7500 kw. at 3600 R.P.M.. It is connected to steam mains at 377 lb./sq. in. (26.5 kg./cm.^{2}) gauge, 545° F. (285° C.), at 109 lb./sq. in. (7.7 kg./cm.^{2}) and at 25 lb./sq. in. (1.75 kg./cm.^{2}) gauge. Each velocity wheel has two groups of nozzles. The live steam valves are on the top of the casing, those for the extraction are at the side of the turbine.

In Fig. 182 may be seen a recent extraction back-pressure turbine for 2500 kw. at 3000 R.P.M.. The initial conditions are 370 lb./sq. in. (26 kg./cm.^{2}) gauge, 700° F. (375° C.), the extraction pressure is 150 lb./sq. in. (10.5 kg./cm.^{2}) gauge, the back-pressure 42.5 lb./sq. in. (3 kg./cm.^{2}) gauge and the maximum extraction 47,500 lb./h. (21,500 kg./h.). The H.P. part has a two-row velocity stage and five impulse stages, the L.P. part has a two-row velocity stage and three impulse stages.

In large factories the requirements for process steam will be known in advance with a considerable degree of accuracy and the choice should preferably fall on a turbine with many stages. Large extraction back-pressure turbines may then often be made with many stages and of the double-casing type (Fig. 183). The *A.E.G.* turbine shown gives 8000 to 14,000 kw. at 3000 R.P.M.. The live steam conditions are 500 lb./sq. in. (35 kg./cm.^{2}) gauge, 750° F. (400° C.), the largest extraction quantity is 110,000 lb./h. (50,000 kg./h.)

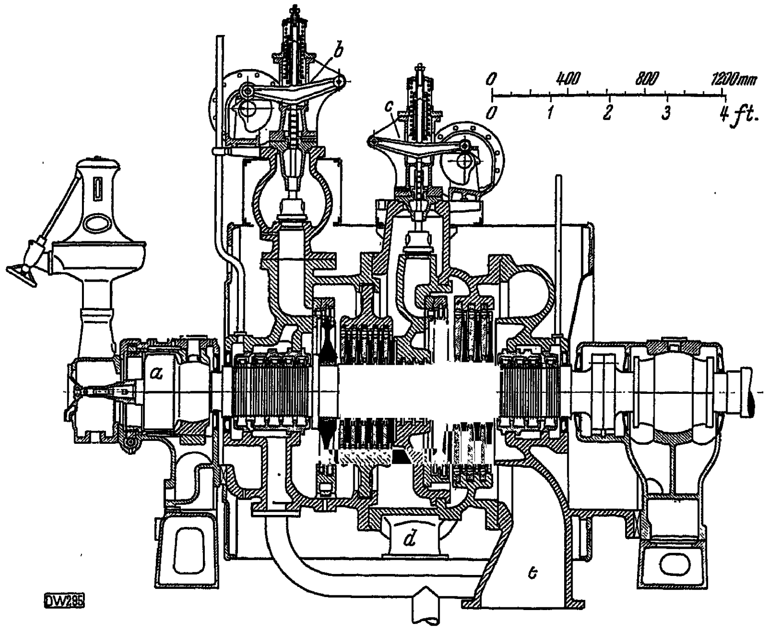


Fig. 182. *A.E.G.*, 2500 kw., 3000 R.P.M., extraction back-pressure turbine

a = Thrust block of the pad type
b = Live steam regulating valve
c = Extraction regulating valve
d = Extraction branch
e = Exhaust branch

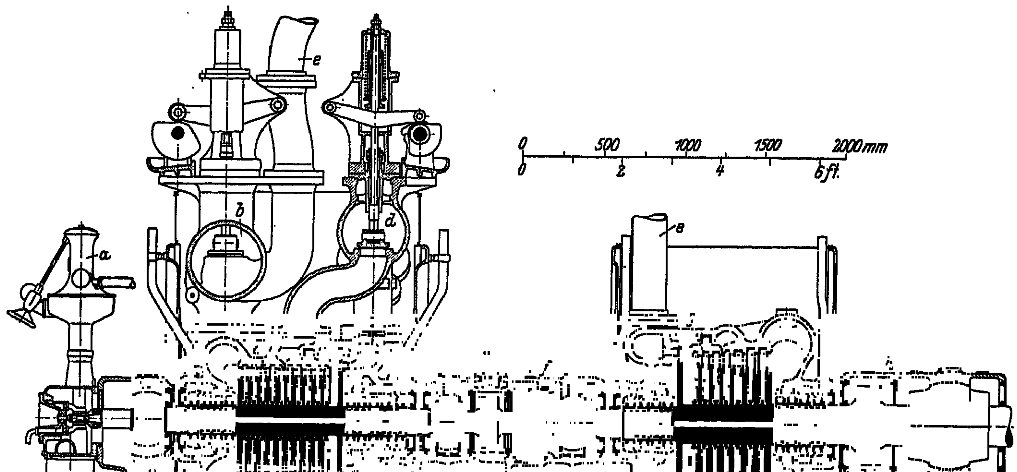


Fig. 183. *A.E.G.*, 14,000 kw., 3000 R.P.M., two-casing, extraction back-pressure turbine

a = Speed governor
b = Extraction regulating valve
c = H.P. casing
d = Live steam regulating valve
e = Pipe connecting H.P. and L.P. casings
f = Thrust block of the pad type
g = L.P. casing
h = Flanged coupling

at 200 lb./sq. in. (14 kg./cm.^{2}) gauge and the back-pressure is 42.5 lb./sq. in. (3 kg./cm.^{2}) gauge. The flows through the H.P. and the L.P. parts are in opposite directions. The two shafts are rigidly coupled and are held in position

by a single thrust block which takes up any unbalanced thrust. Both rotors have a number of impulse stages, the discs being cut from the solid.

This type of design was also adopted for the machine which is probably the largest extraction back-pressure turbine ever built. It is for 20,000 kw. with working conditions of 484 lb./sq. in. (34 kg./cm.²) gauge, 750° F. (400° C.), 243,000 lb./h. (110,000 kg./h.) extraction at 170 lb./sq. in. (12 kg./cm.²) gauge and 50 lb./sq. in. (3.5 kg./cm.²) gauge back-pressure. The largest amount of steam flowing through the H.P. part at 20,000 kw. load with normal steam conditions is no less than 574,000 lb./h. (260,000 kg./h.) whilst with reduced live steam conditions the maximum quantity will be 617,000 lb./h. (280,000 kg./h.).

Fig. 184 shows a machine which has been designed throughout as a multistage reaction turbine. It gives 4600 kw. at 3600 R.P.M. with initial steam conditions of 150 lb./sq. in. (10.6 kg./cm.²) gauge and about 465° F. (240 ° C.). The steam which is bled after the second and third group of stages and the whole of the exhaust steam go to large feed-water heaters. The machine is a house set

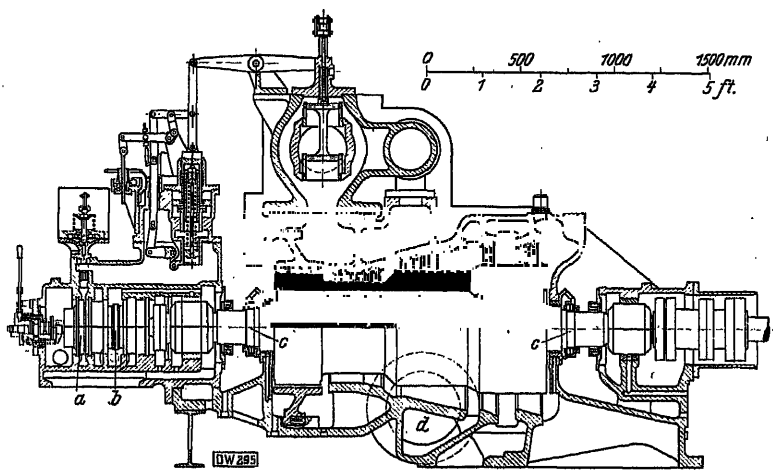


Fig. 184. *Westinghouse,* 4600 kw., 3600 R.P.M., extraction back-pressure turbine
a = Oil impeller for governing b = Thrust bearing of the pad type c = Water-sealed gland d = Extraction branch

in a large power station. Regulation is by a speed governor and a pressure regulator at the first extraction point.

In order to obtain the required load, an extraction back-pressure turbine may also be joined to a condensing part which will then have to deal with more or less L.P. steam according to the load. The machine obtained will be an ordinary extraction turbine with two extraction points, but it may also be called an extraction back-pressure turbine with a condensing part. There is, naturally, no fundamental reason for not further increasing the number of extraction points; it is not always advisable, however, as the plant becomes too complicated and the efficiencies diminish greatly when the load and extractions vary. A better solution would be to provide two or more units.

d. Low-pressure and mixed-pressure turbines

A low-pressure turbine may be considered to a certain extent an inversion of a back-pressure turbine. In the latter case the important factor will be the requirements of process steam at a given pressure. This steam will be generated at a higher pressure and will work in a turbine between the boiler and the process main. The case of the low-pressure turbine will be exactly the opposite; exhaust steam from hammers, presses and similar non-condensing machines will be used at about atmospheric pressure. The heat drop will be increased upwards

for back-pressure turbines and downwards for low-pressure turbines. In neither case can this be done without expense, it will require a small additional quantity of fuel or some power for the condenser auxiliaries. There will be a definite relation between output and steam quantity in the case of pure low-pressure turbines also. It will only be possible to use them when they can be connected in parallel with other generating units. For this reason, they are not often employed, and when they are it is always in conjunction with steam accumulators. Mixed-pressure turbines are more often used. They consist of a low-pressure condensing turbine to which has been added a H.P. part for obtaining any additional power required. Machines of this kind are often used in steel works where they usually drive compressors or blowers.

A mixed-pressure turbine of recent design is illustrated in Fig. 185. It is for 4000 kw. at 4000 R.P.M.. The H.P. part has a two-row velocity stage and two single-row impulse wheels; the L.P. part has two discs also, the first one working only with low-pressure steam. The turbine can take steam at three different pressures. The H.P. part has two separate steam belts, one for live steam at 107 lb./sq. in. (7.5 kg./cm.^{2}) gauge, the other for I.P. steam at 64 lb./sq. in. (4.5 kg./cm.^{2}) gauge. After the H.P. and the I.P. steam has worked in the H.P. part it passes round the first L.P. stage to the last stage, where it mixes with the L.P. steam. A design of this type, with separate belts for the H.P. and L.P. steam, has several advantages over designs having the ordinary method of admitting the L. P. steam. If all the steam were admitted together in the L.P. part, at partial loads, when the H.P. and I.P. steam quantities are small, the efficiency of the H.P. part would decrease through an excessive leaving loss, whilst the L.P. steam would be throttled in the admission. The machine would be considerably less economical than if it had separate admission for the L.P. steam. The H.P. turbine expands steam down to the low pressure which is to be found before the last L.P. stage. Thus, when the turbine is working on L.P. steam only, it is possible to avoid the H.P. part running idle in a pressure which would be at least atmospheric.

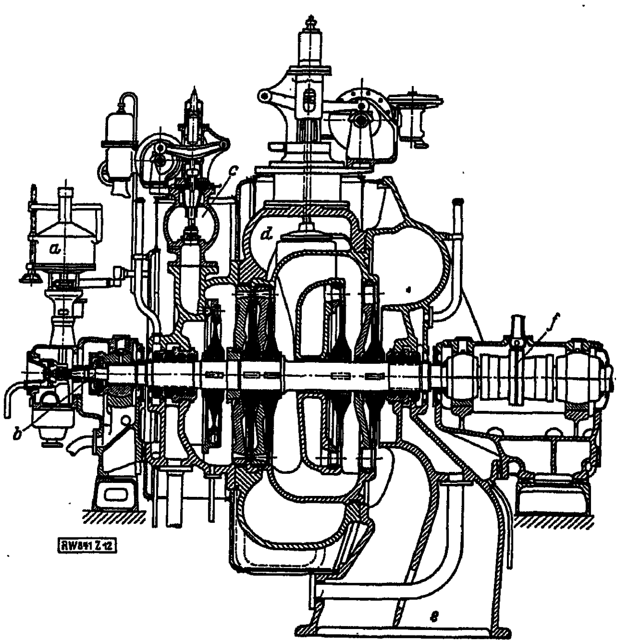


Fig. 185. *A.E.G.*, 4000 kw., 4000 R.P.M., multiple-pressure low-pressure turbine for driving fan

a = Speed governor
b = Thrust block of the pad type
c = Regulating valves for H.P. and for I.P. steam
d = Regulating valve for L.P. steam
e = Exhaust branch
f = Double toothed type coupling

Another type of mixed-pressure turbine is given in Fig. 186. For live steam pressures between 140 and 200 lb./sq. in. (10 and 14 kg./cm.^{2}) gauge, *B.B.C.* build machines of this type with a velocity wheel in the H.P. part and pure reaction blading in the L.P. part. The admission valves for the L.P. steam are on either side of the turbine.

The governing of mixed-pressure turbines is generally arranged so that the machine will use all the L.P. steam available and H.P. steam will only be em-

ployed to make up any deficiency in output. The speed governor will regulate the output by acting on the H.P. and L.P. relays in the same direction. The operation will take place so that when the load increases, for instance, the L.P. valve will be opened first, then the H.P. valve; whilst the sequence will be reversed for a decrease in load, the H.P. valve being closed first. The pressure regulator of the L.P. steam alters the division of the load between the two steam supplies according to the quantity of L.P. steam available. It will act in opposite directions on the H.P. and L.P. gears; thus, when the available quality of L.P. steam increases, the L.P. valves will be opened and the H.P. valve closed. The opening and closing will be performed in such a way that the output does not vary. A device is usually provided on the H.P. gear which alters the leverage between the two valves in accordance with the live steam pressure. In this way, when the L.P. regulator operates the output will not vary, whatever may be the live steam pressure.

Accumulator turbines have developed from mixed-pressure turbines and are at the present time becoming increasingly popular. The *Rateau* type of accumu-

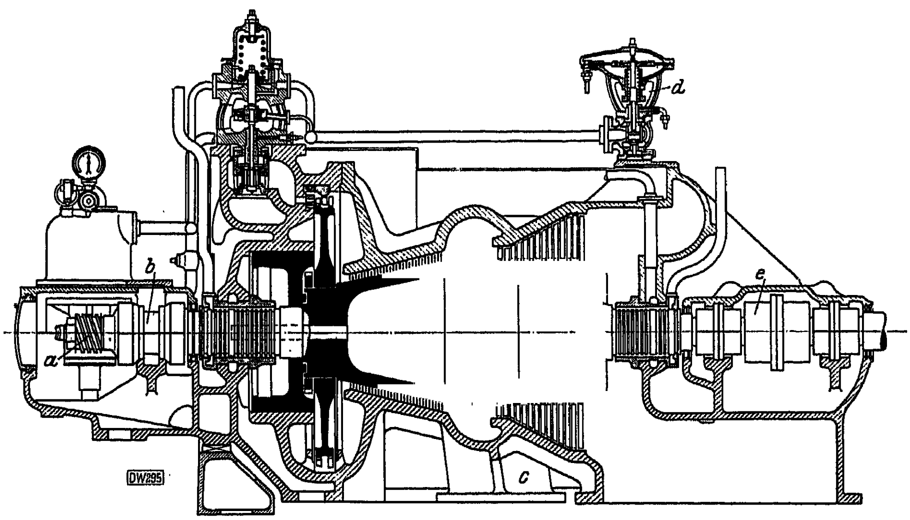


Fig. 186. *B.B.C.*, mixed-pressure turbine for live steam and low-pressure steam

a = Governor drive
b = Journal bearing with double thrust bearing of the pad type
c = Branch for admitting L.P. steam
d = Pressure governor
e = Double claw-type coupling

lator will only deal with small variations in pressure. Hence, a turbine drawing steam from an accumulator of this kind will work under almost the same conditions as a low-pressure turbine supplied with exhaust steam at a pressure which is nearly constant. On account of this small variation in pressure, *Rateau* accumulators have only a limited storage capacity and have not gained a wide application.

The *Ruths* accumulator earned its popularity because it worked with wide variations in pressure and had a considerable capacity. It made it necessary, however, to develop a special type of turbine which would be suited for working with greatly fluctuating steam pressures. There were, thus, two problems to solve. In the first place, the turbine had to be designed for varying steam quantities and had to have as constant an efficiency as possible. In the second place, the governing had to be adapted to the special conditions which usually occur when live and accumulator steam are used. According to the practical results which are available at the present time it would appear that both difficulties have been entirely overcome in several turbine designs. Thus, accumulator

turbines are well on the way to becoming an individual type, like back-pressure or extraction turbines.

The essential principle of the governing of accumulator turbines is that as the accumulator pressure sinks additional valves will automatically be opened for maintaining the load, also pressure regulators will change the method of operation so that the turbine may either run on H.P. or L.P. steam. This subject requires to be mentioned here only as two examples have previously been given on pages 68 to 70.

Accumulator turbines have to be designed so that over the largest possible range of loads the efficiency will remain as nearly constant as possible. This can best be accomplished by not having too many stages. Fig. 187 shows a turbine for live and accumulator steam which is for supplying power for electric

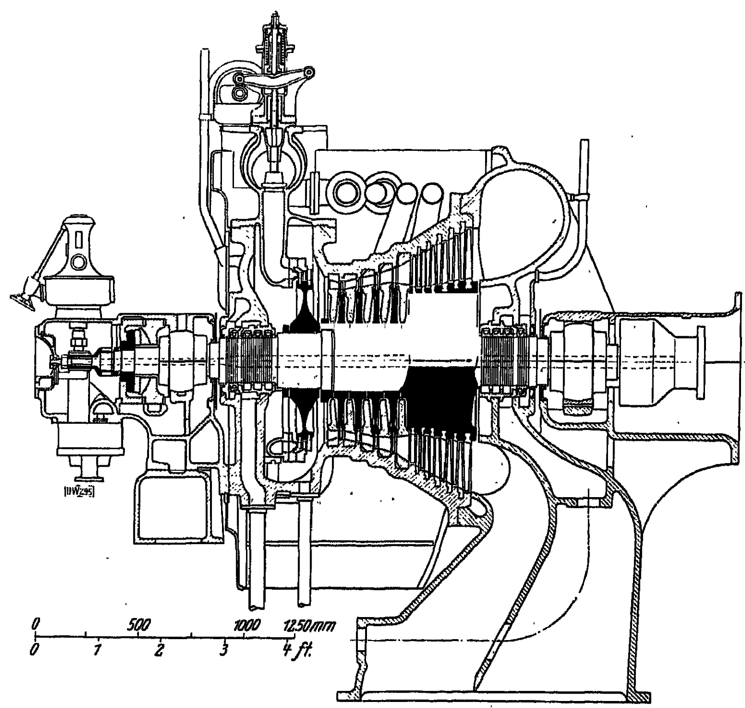


Fig. 187. *A.E.G.*, 10,000 to 15,000 kw., 3000/1000 R.P.M., geared condensing turbine for operating with live steam and accumulator steam

traction. It must be capable, therefore, of meeting particularly large peak loads of short duration by means of accumulator steam. The normal live steam conditions are 200 lb./sq. in. (14 kg./cm.²) gauge, 680° F. (360° C.) and the accumulator pressure varies between 215 and 43 lb./sq. in. (15 and 3 kg./cm.²) gauge. The normal output of the turbine is 10,000 kw.. Overloads of 25% can be obtained during fifteen minutes at intervals of one hour and 15,000 kw., or 50% overload, can be obtained during three minutes at intervals of fifteen minutes. A two-row velocity stage is provided for regulation, live steam being admitted round half the circumference and accumulator steam round the other. This stage is followed by five impulse and five reaction stages. The machine drives a single-phase alternator running at 1000 R.P.M. and producing 16⅔-cycle current for traction. The turbine is for 3000 R.P.M. and it is coupled to the alternator through gears, details of which are given in Fig. 208.

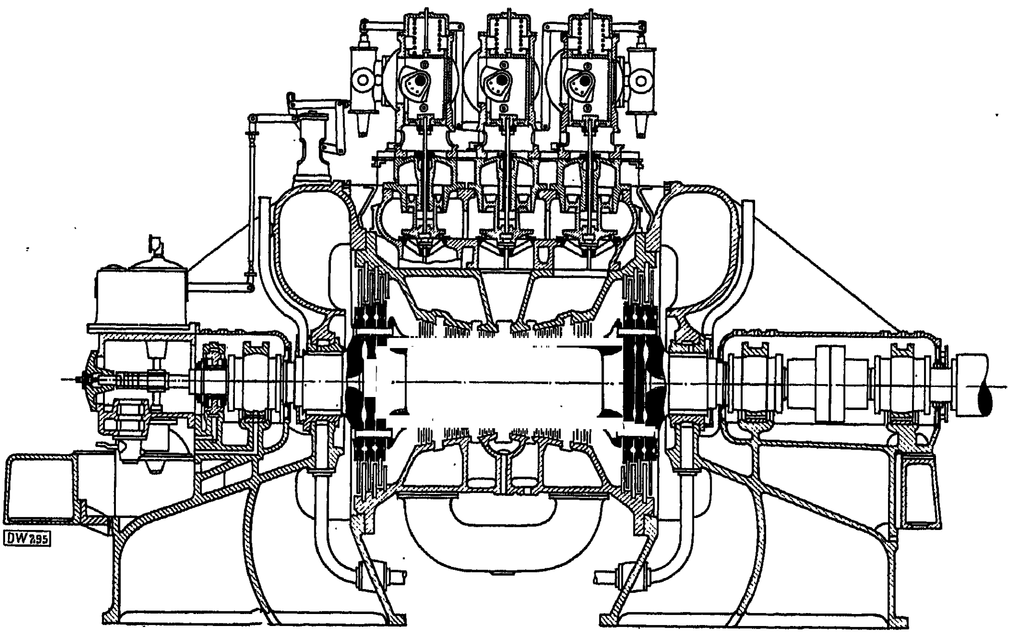


Fig. 188. *S.S.W.*, 20,000 kw., 3000 R.P.M., single-casing, double-flow accumulator turbine

Fig. 188 illustrates a turbine of *S.S.W.*'s make running on accumulator steam only. It gives 20,000 kw. at 3000 R.P.M. and was built for the Charlottenburg Power Station, which has the largest accumulator plant yet installed for power production. The turbine has two flows with 16 stages each, the four first stages being by-passed when the accumulator pressure falls. It may be noted that the turbine has throttle governing only.

e. Special designs

The principal designs of turbines for an axial steam flow have been amply described in the previous pages. By far the largest number of turbines being of this type, it was not considered necessary to adopt the logical method of classification, which would have been to have compared axial and radial-flow turbines in every chapter. Some types of turbines not having an axial flow maintain their position over a certain range of output and have acquired some importance. For these reasons the subject is treated briefly under a separate heading. The types which will be described are the double rotation *Ljungström* turbine with a radial flow in the outwards direction, the *Elektra* turbine with a radial flow in an inwards direction, finally, the *Terry* turbine, which is extensively used in America and is similar to the former *Riedler-Stumpf* turbine.

The *Ljungström* turbine has two discs revolving in opposite directions. They both carry rows of blades which penetrate between each other. The steam flows through the blading from the centre towards the circumference. Each disc is on an overhung shaft and a *Ljungström* turbine will require two generators each for half the total output (Fig. 189). Thus, there are two fundamental differences between this type of turbine and the ordinary axial type: the radial steam flow and the opposite directions of rotation.

The first consequence of radial flow will be a very short overall length of the turbine as the stages are not arranged in an axial sequence but fit between each other on either side of a plane of symmetry. The increase in length due to

the two generators must also be considered, and a shorter turbo-generator set will usually be obtained when an axial-flow turbine in a single casing is used; if, however, a multi-casing design is required it will have a greater length. Radial-flow turbines will be more closely limited in size by the greatest blade length which is permissible and they are, therefore, best suited for small and medium outputs.

As a result of the two directions of rotation a good efficiency will be obtained with a relatively small number of stages. It is not possible, however, to consider the theoretical aspect of the question here. *Stodola* devotes a whole chapter to this type of turbine (61).

Fig. 190 shows the sectional arrangement of a modern *Ljungström* turbine for a maximum load of 14,000 kw. at 3000 R.P.M.. Overhung at the end of both shafts are the discs which rotate in opposite directions and carry the blading. Live steam is admitted through a number of pipes *c* and holes *d* in the hubs of the discs allow it to reach the central space immediately in front of the first stage. It then flows outwards in a main radial direction through the different stages until its volume becomes too large for an axial blade. The two last stages have to be for an axial flow. They are on separate discs, similar pairs

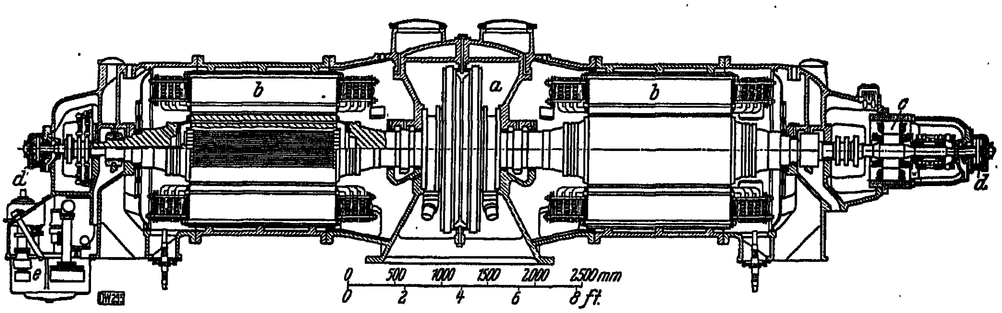


Fig. 189. 14,000 kw., 3000 R.P.M. *Ljungström* turbo-alternator

a = Turbine, 14,000 kw.
b = Alternator, 7000 kw.
c = Exciter
d = Single-collar thrust bearing with pivoted pads
e = Oil tank

being provided on either side of the radial-flow blading. The expanded steam leaving the last stage is led to the condenser by way of the exhaust belt, which is formed by the entire casing in *Ljungström* turbines.

Each rotating disc consists of several parts, in the first place the disc proper, towards the circumference of which are fixed the wheels of the axial stages. The manner in which the different parts of the rotor are assembled prove the great technical abilities of the inventor. They are all held together by conical bolts and firm but not entirely rigid connections are obtained. In this way, a sufficient degree of reliability is acquired, a very accurate manufacture being essential for this type of turbine. The principle of providing joints, which enable a certain flexibility to be obtained for heat or mechanical deformations in rotating parts, has been applied also for attaching the blade rings to the discs. For this purpose rings of dumb-bell section are rolled into the disc and the blade rings. The gap between the rings of blades is sealed by thin strips of nickel or nickel steel at the end of the blades having 0.004 in. (1/10 mm.) clearances.

When the steam flow is radial the leakage losses in clearances and glands are completely altered. In the first place, the usual outer gland which seals the vacuum against atmospheric pressure is eliminated and there is no possibility

(61) 6th (German) Edition p. 250; English Edition p. 300.

of air leaking into the exhaust casing. On the other hand, two glands are required for withstanding the full pressure after the throttle valve, which may sometimes be equal to that of the live steam. *Ljungström* designed for this purpose a special type of gland of the labyrinth type with numerous constrictions and with overlapping edges. In this way he was able to provide in an axial length of hardly 4 in. (100 mm.) a great number of constrictions, 80 to 100 being often used. A second feature not found in axial turbines is the inner packing gland for separating the space at live steam pressure from the exhaust belt. The leakage takes place between the rotating discs and the casing and not through the blade clearances. Thus, a second gland will be necessary for sealing the live steam pressure against the vacuum; its design is given in Fig. 191. The dimen-

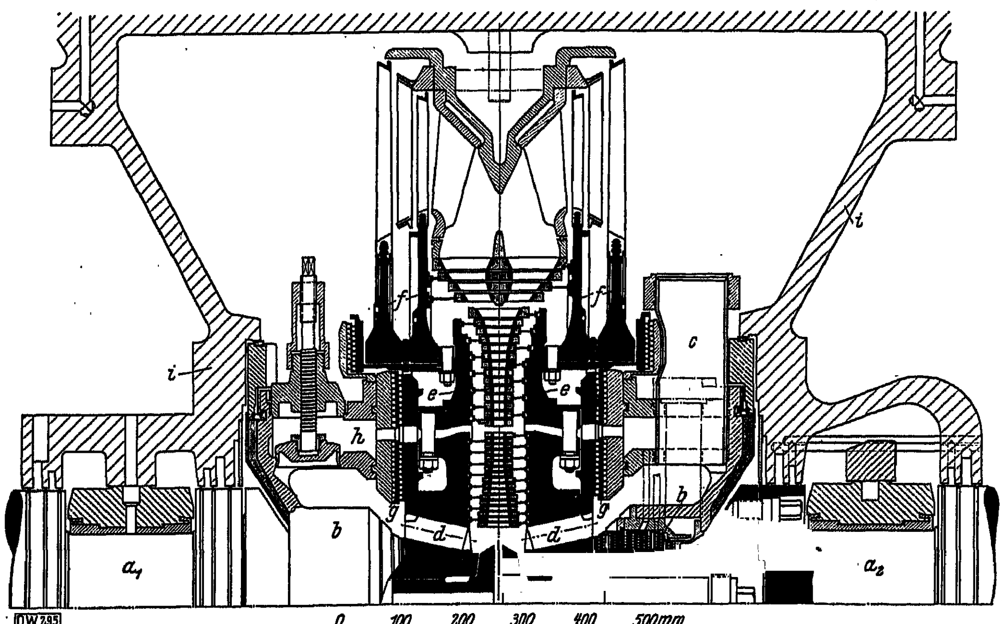


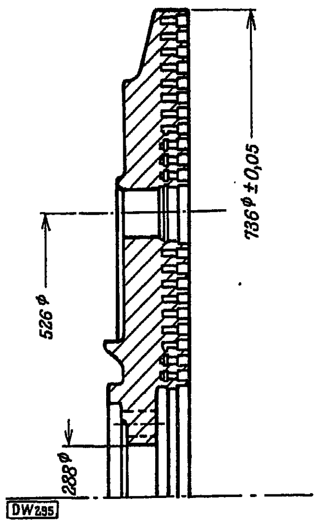
Fig. 190. 14,000 kw., 3000 R.P.M. *Ljungström* turbine

$a_{1,2}$ = Turbine shaft
b = Packing gland
c = Live steam supply
d = Holes through the rotating discs for admitting the live steam
e = Rotating disc carrying the radial-flow blading
f = Rotating disc carrying the blading of the last radial-flow stages and of the axial-flow stages
g = Inner packing gland between rotor and casing
h = By-pass
i = Casing

sions and tolerances given on this drawing show what great accuracy is required in the manufacture.

The casing of a *Ljungström* turbine is illustrated in Fig. 192. The two flanges at the ends are bolted to the generator and the weight of the entire set is transmitted to the condenser through the exhaust flange. In the case of larger machines the alternators have also light supporting columns. The price of the foundations will be reduced with these machines, therefore, as no special constructions are required level with the turbine.

Ljungström turbines up to about 15,000 kw. have been built and are in service. Very small steam consumptions have been obtained, for sets of small capacities especially, and in this respect they are somewhat better than axial turbines. At the same time they can be very quickly started up, five to eight minutes being required from cold. The turbine has these advantages together with a small initial cost, but the design may appear rather bold and hazardous,



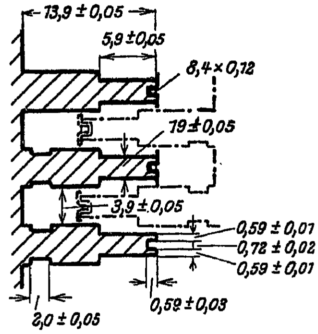


Fig. 191. Rotating disc of the radial-flow inner packing of a *Ljungström* turbine (dimensions in mm.)

although it is ingenious, and it will be sensitive to dirt. At the present time a 23,000 kw. unit (27,000 kw. maximum) and one for 30,000 kw. (50,000 kw. maximum) are under construction and they will be the largest sets existing.

As a second example of a radial-flow machine the *Elektra* turbine of the German firm of *Kühnle, Kopp & Kausch A. G.* will be mentioned. It was invented by *Kolb* and has acquired a certain importance. The steam on leaving the stationary nozzles flows inwards in a combined tangential and radial direction and does work in several velocity stages, a corresponding number of reversals in the direction of the jet being required. These turbines do not possess the quality of having double rotation, also several stages in series do not have such a smooth steam path as either axial-flow or *Ljungström* turbines and the design is chiefly used for small units.

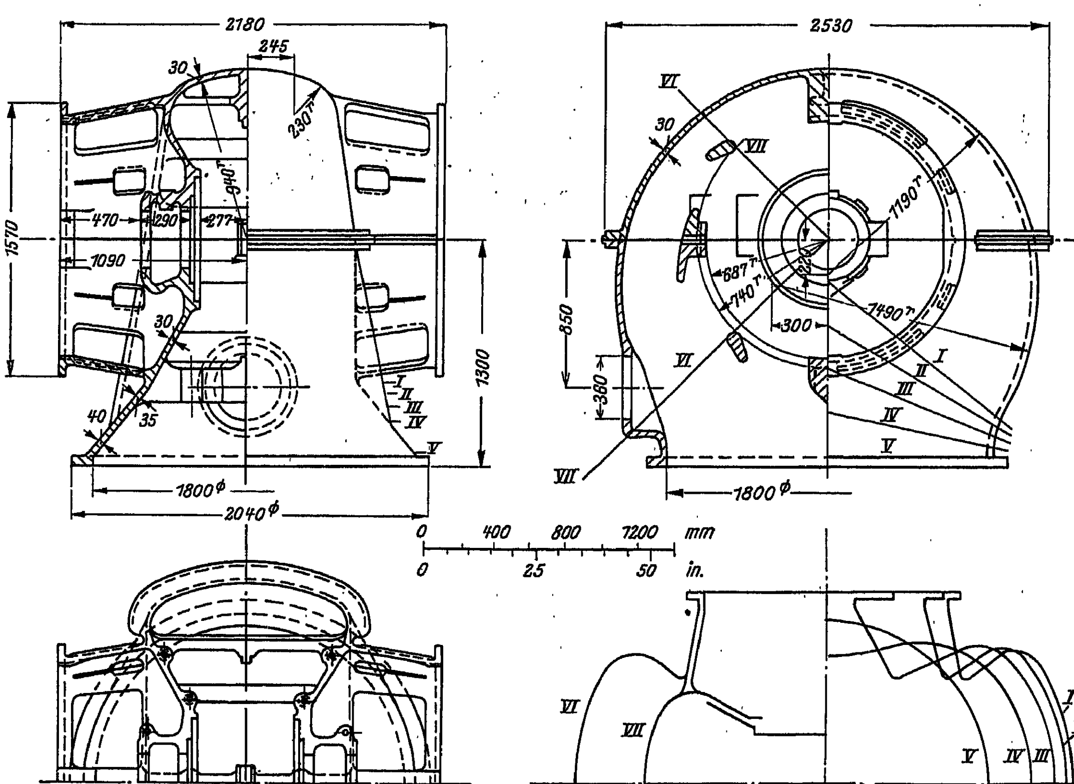


Fig. 192. Casing of a *Ljungström* turbine

Fig. 193 shows the sectional arrangement of a two-stage *Elektra* turbine. As may be seen, the steam enters the axial blades from the outside and is reversed once or several times on having passed through them. Hence, the turbine is of the impulse type with velocity stages. The turbine shown is for high steam pressures.

The method used for reversing the jet may plainly be seen in Fig. 194. According to the output required, one or several nozzles are provided around the circumference to give sufficient admission. Special methods are employed for fixing the moving blades, the wheels are surrounded by a circle of blades and distance pieces over which a strong ring is shrunk. Some types have two rows of blades, one on each side of the disc. The shafts always run below the critical speed. The glands are of the carbon-ring type.

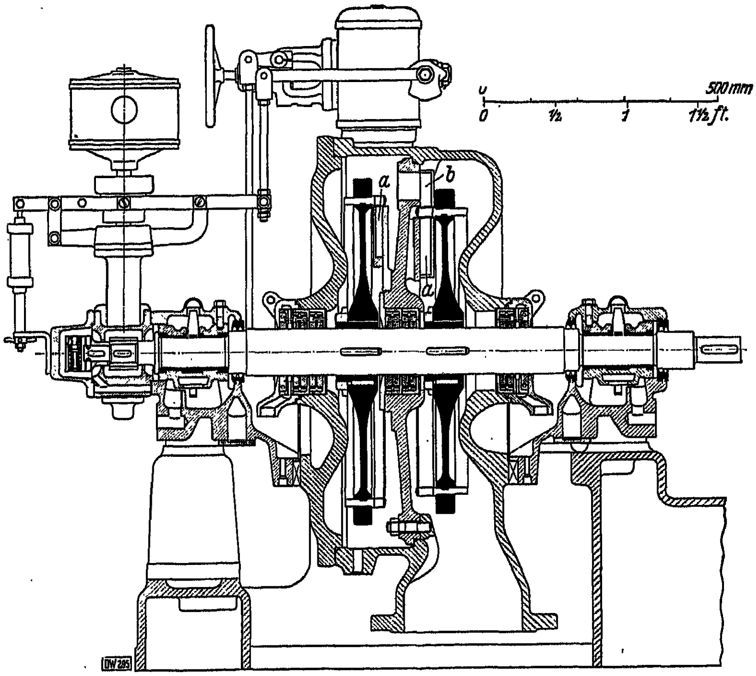


Fig. 193. *Kühnle, Kopp & Kausch,* two-stage back-pressure turbine of the *Elektra* type

a = Reversing chamber b = Nozzle of the second stage

A design with admission in a combined radial and axial direction is executed in America by *Terry*. Only back-pressure units of not more than about 300 kw. are built. The efficiencies are not equal to those of axial-flow turbines and it is mainly for its originality that the design is of interest.

It may be seen from Fig. 195 that the *Terry* turbine works as a multiple-row velocity wheel. The jet issuing from the nozzles is reversed 180° in the moving buckets, it enters a stationary reversing chamber and is returned to the wheel, the process being repeated several times. The steam path will have a helical form. The wheel is in one piece, the buckets being milled by means of a special cutter.

The sectional arrangement of a modern turbine of this type is illustrated in Fig. 196. It gives about 300 kw. at 1465 R.P.M. and uses steam at about 400 lb./sq. in. (28 kg./cm.²) gauge, 750° F. (400° C.), the back-pressure being 3 lb./sq. in. (0.21 kg./cm.²) gauge. It drives a high-pressure pump. The wheel

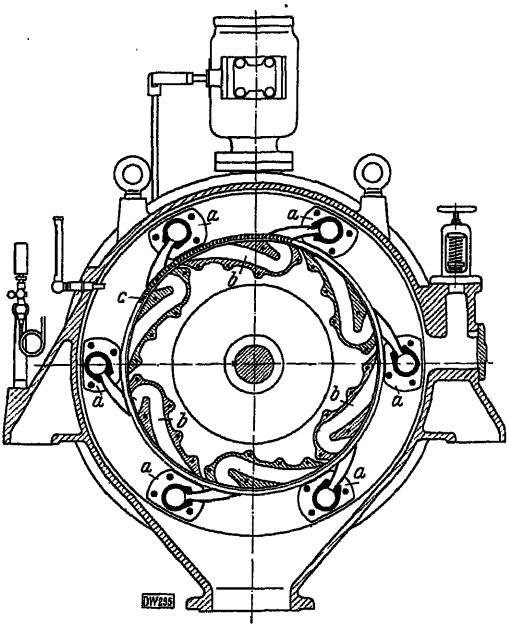


Fig. 194. *Kühnle, Kopp & Kausch,* cross-section through a single-stage *Elektra* turbine for 350 to 500 kw.

a = Live steam nozzle b = Reversing chamber
c = Rotating blades

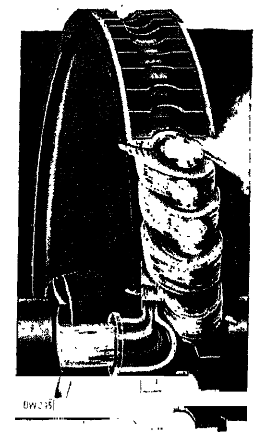


Fig. 195. Steam path in a *Terry* turbine with a multiple-row velocity stage

is fixed on the shaft by means of two nuts and a key. The casing has a horizontal joint only. To meet the varying needs of steam pressure and temperature the turbine rests either on strong feet on the casing or on bearing pedestals so as to allow sufficient latitude for heat expansion. In the first case, which is for low steam conditions, the bearings are cast with the lower half of the casing, whilst in the second case they are bolted on as in Fig. 196.

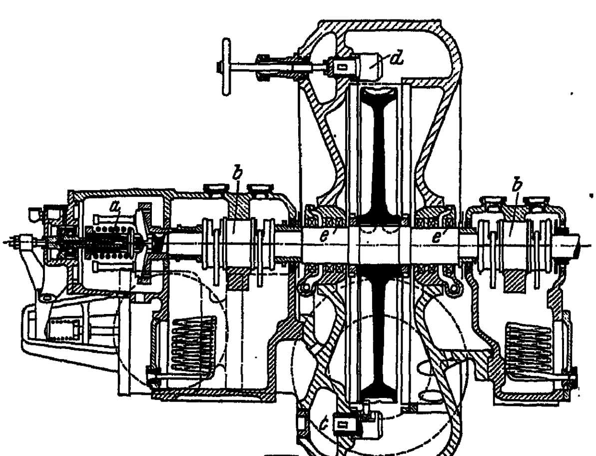


Fig. 196. *Terry,* 300 kw., 1465 R.P.M. back-pressure turbine

a = Speed governor d = Nozzle regulated by hand
b = Bearing with ring lubrication e = Carbon-type packing gland
c = Nozzle controlled by governor

Carbon-type packing glands are used. The bearings are long owing to the ring-type lubrication and the bearing pressure is kept very low. A separate thrust bearing is not provided, but only collars on either side of the governor end bearing. These are quite sufficient as only small thrusts will occur. Each bearing pedestal is also an oil tank and has a spiral cooling pipe. The centrifugal speed governor is mounted directly on the end of the shaft and operates the throttle valve by means of a lever, no relay being employed. The overload valves are hand operated. A special emergency valve is placed before the throttle valve.

This turbine owes its great popularity in America to its main quality, its reliability. The makers lay great stress on the fundamental rule, that even when a cheap machine is required, the safety factor must never be reduced to the minimum. Apart from this point, the design itself has a quality which is often so much sought after in America that it may compensate for a low efficiency; it is the large clearances between the stationary and rotating parts (Fig. 197). The radial play is between the rims of the wheel and the casing or the edges of the reversing chamber and any rubbing would not damage the blades. The side clearance is very large, being about one inch (25 mm.), and even a thrust from the driven machine cannot cause any injury to the blading.

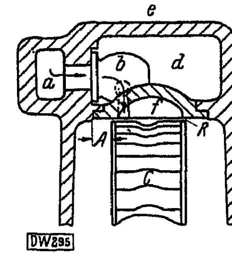


Fig. 197. Radial (R) and axial (A) clearances between rotor and casing of a *Terry* turbine

a = Steam belt
b = Nozzle
c = Wheel
d = Exhaust chamber
e = Turbine casing
f = Reversing chamber

In conclusion, the *Terry* turbine is another example showing that up to the present axial turbines are more practical and have better efficiencies than any other type. However efficient may be the *Ljungström* turbine, however compact the *Elektra* turbine and however reliable the *Terry* turbine, each for its own range of output, when a combination of all these qualities is required, they were up to the present not able to dispute the leading position of axial turbines, now so highly developed after years of experience.

2. Turbines for indirect drive

It may not be possible to find a sufficiently economical speed of rotation by taking an average value between the best speed for the turbine and the best speed for the driven machine. A speed reducer should then be employed. Gears are generally used; other methods, such as *Föttinger*'s hydraulic transformer or an electric coupling by means of a generator and motor, have found but a limited application only for ship propulsion.

If a suitable oil is employed the friction losses in gears for moderate outputs, inclusive of the bearing losses, are 1 to 2% of the power in the case of single reduction or 3 to 4% in the case of double reduction. The more economical running of the driving and driven machines compensates for these small losses. It must be remembered, however, in the case of small turbines especially, that an increase in speed will result in greater mechanical losses in the turbine and its auxiliary machinery, such as governor or oil pump drives. It may be thought that a plant with a geared turbine is not so reliable as one for direct drive. These scruples are no longer justified in view of the advanced knowledge we now possess concerning the performance of gear wheels and of having overcome the difficulties of finding suitable materials, satisfactory design and manufacturing methods. The reliability will even be greater for higher speeds as the dimensions and weights will be less; thus, the pressure and temperature stresses will be diminished and the rotating masses reduced. Another advantage will be the smaller size and weight of the set. Naturally, gears should only be used when they considerably improve the operation of the whole plant.

When the available heat drop and the steam quantity are small, as in the case of small back-pressure turbines especially, a single-row impulse wheel or a two-row velocity wheel will be appropriate. The turbine will run at a high speed and will have partial admission. Fig. 198 shows a turbine with a single-wheel working as a velocity stage. There is only one row of moving blades. On having passed through the blading the jet is turned in a reversing chamber and led once more to the wheel. The pinion is supported by two large bearings

and the turbine disc is overhung at the end of the pinion shaft. The usual coupling between turbine and gears is not required and only one packing gland is necessary, points which are of especial importance in turbines for high

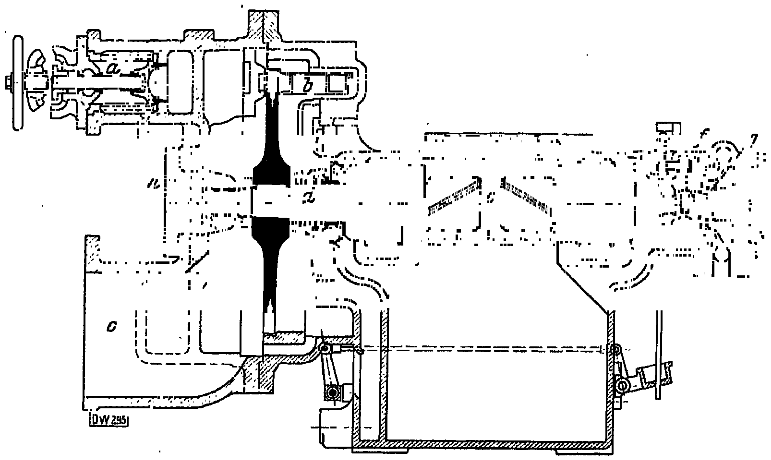


Fig. 198. *Westinghouse,* 75 to 500 kw., 8400 to 5000 R.P.M., single-stage, geared back-pressure turbine with reversing chambers

a = Throttle valve
b = Reversing chamber
c = Exhaust branch
d = Water-sealed gland
e = Pinion
f = Emergency governor
g = Oil impeller for governing
h = Restraining socket

pressures. The same method of governing is employed as in large *Westinghouse* turbines, a centrifugal oil pump being provided at the end of the shaft (refer to pages 75 to 76). Normal speeds for this type of turbine are between 8400 and 5000 R.P.M. and, according to the conditions, the output will be between 75 and 500 kw..

The *A.E.G.* also make a single-stage back-pressure turbine for outputs up to about 500 kw.. It is provided, however, with a two-row Curtis wheel (Fig. 199). It is suited for heat drops of between about 63 and 216 B.Th.U./lb. (35 and 120 kcal./kg.). For these conditions the ratio c_0/u of the steam to the blade velocity may be chosen between about 3.8 and 6 for two-row velocity wheels or 2.3 to 3 for single-row impulse wheels; it may even be possible to limit these values still further. The great advantages of a turbine of

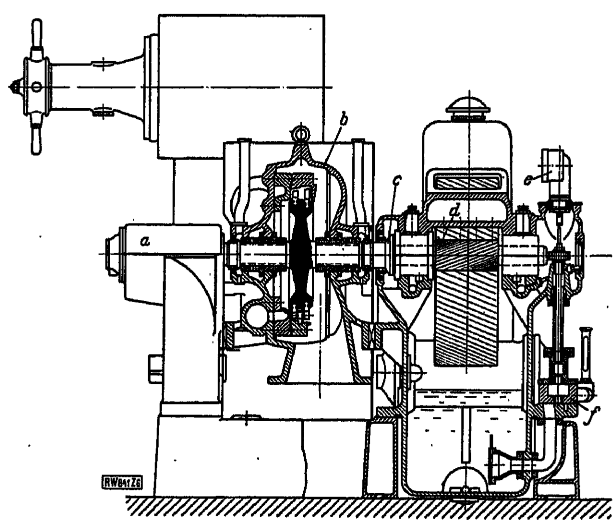


Fig. 199. *A.E.G.,* 500 kw.$_{max.}$, 7500/1000 R.P.M. small geared turbine

a = Speed governor
b = Turbine
c = Thrust bearing
d = Gears
e = Tachometer
f = Oil pump

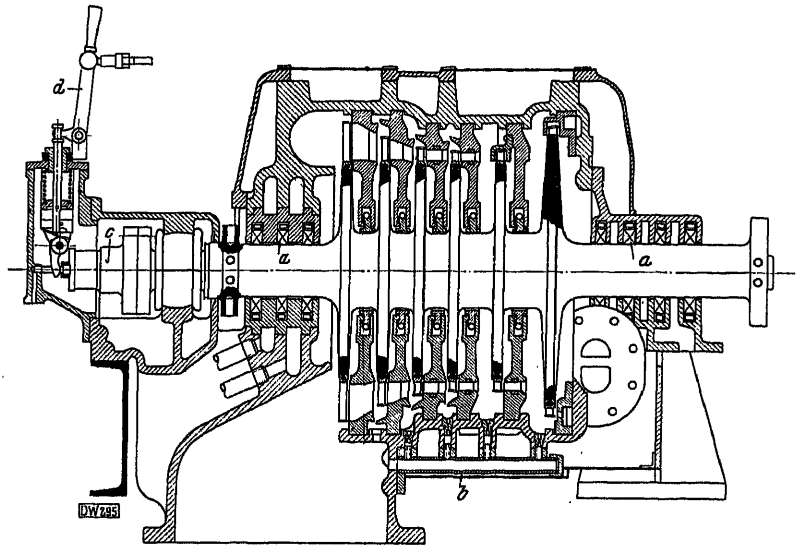


Fig. 200. *Amer. G.E.C.*, 250 kw., 10,000 R.P.M., geared high-pressure turbine

a = Carbon-type packing gland c = Speed governor
b = Drain d = Hand trip lever

this type are that good efficiencies are maintained for partial loads and overload and as a result of the large blade clearances the machine is very reliable and can quickly be started up.

Most multi-stage turbines for small or moderate outputs (up to about 2000 kw.) are of the impulse type with discs, and large blade clearances are possible. As examples two small turbines of the *Amer. G.E.C.* are shown in Figs. 200 and 201. The first machine is for 285 lb./sq. in. (20 kg./cm.^{2}) gauge, 700° F. (370° C.) and about 0.85 lb./sq. in. (0.06 kg./cm.^{2}) absolute back-pressure. It gives 250 kw. at 10,000 R.P.M.. The rotor is cut out of one piece. Interesting details are the carbon-type glands and the diaphragms which have obviously been designed to ensure efficient drainage. The second turbine has four stages and

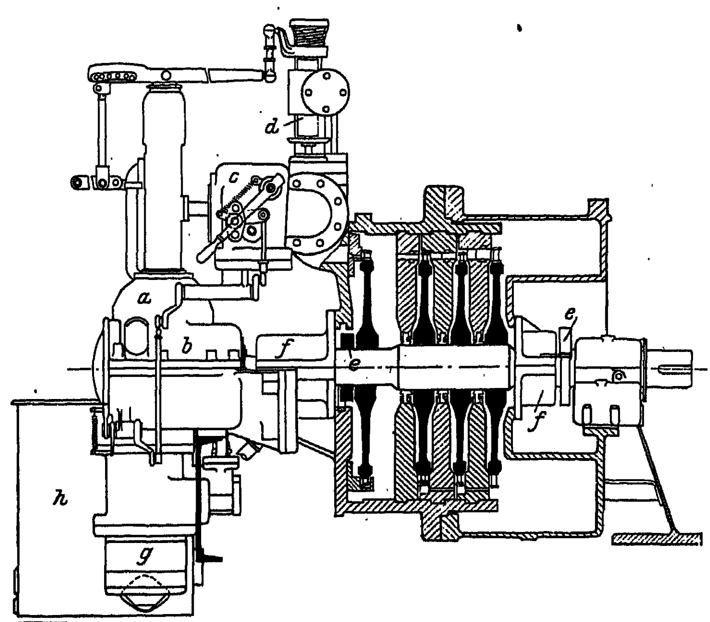


Fig. 201. *Amer. G.E.C.*, 750 kw., 7700 R.P.M., geared back-pressure turbine

a = Speed governor e = Balancing ring
b = Thrust block f = Packing gland
c = Chest of emergency valve g = Oil pump
d = Throttle valve h = Oil tank

is for 390 lb./sq. in. (27.5 kg./cm.^{2}) gauge, 700° F. (370° C.) and a moderate back-pressure. It has an output of about 750 kw. at 7700 R.P.M.. The casing may be noticed as it is entirely made of welded steel plating.

Fig. 202 illustrates a small *Parsons* high-pressure turbine of 100 kw. at 10,000 R.P.M.. It has a two-row impulse wheel with partial admission and five plain impulse wheels. The development of steam turbines was very much hindered in its early stages through damage caused by insufficient blade clearances.

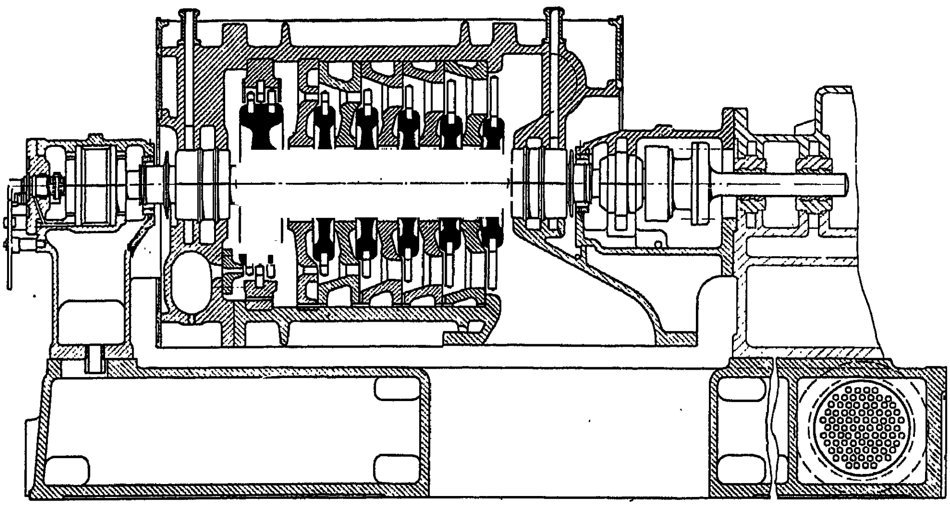


Fig. 202. *Parsons*, 100 kw., 10,000 R.P.M., geared high-pressure turbine

Parsons has not forgotten this since he uses a plain impulse blading for small outputs. This is the only method of obtaining absolute reliability in such cases owing to the small dimensions and steam quantities.

The American *de Laval Steam Turbine Co.* build an impulse turbine for about 250 kw. and running at 10,000 R.P.M. also (Fig. 203). The original design of its rotor may be noticed. It is coupled through a double helical gear to a generator running at 1200 R.P.M.. An ungoverned extraction branch is provided after the second stage.

B.T.H., also, make small geared turbines of the impulse type (Fig. 204). The turbine shown runs at 8000 R.P.M. and gives about 600 kw.. It drives a D.C. generator and an alternator running at 912 R.P.M., both machines being of about half the total capacity. Attention may be drawn to the flexible coupling with a single set of claws and a corrugated sleeve for increasing the flexibility, and to the three-bearing pinion.

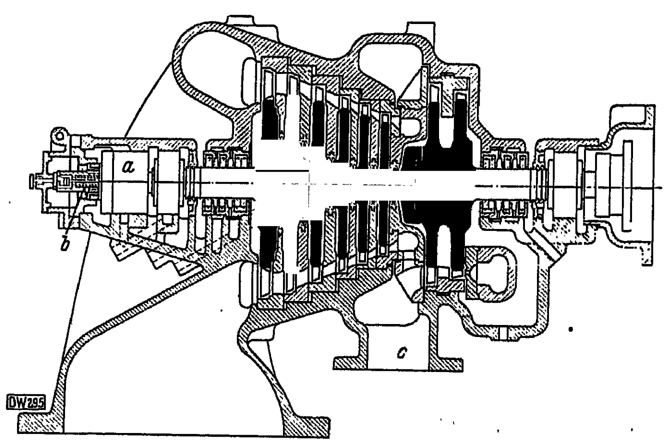


Fig. 203. *Amer. de Laval*, 250 kw., 10,000/1200 R.P.M., geared high-pressure turbine

a = Thrust block b = Emergency governor c = Extraction branch

Fig. 205 gives an example of an *Erste Brünner* geared turbine. It is a single-casing extraction turbine for 1000 kw. at 6000/1500 R.P.M.. Both parts of the turbine consist of impulse wheels and they each have a Curtis wheel with a diaphragm between the two rows of moving blades. The coupling is of an original design in this example also.

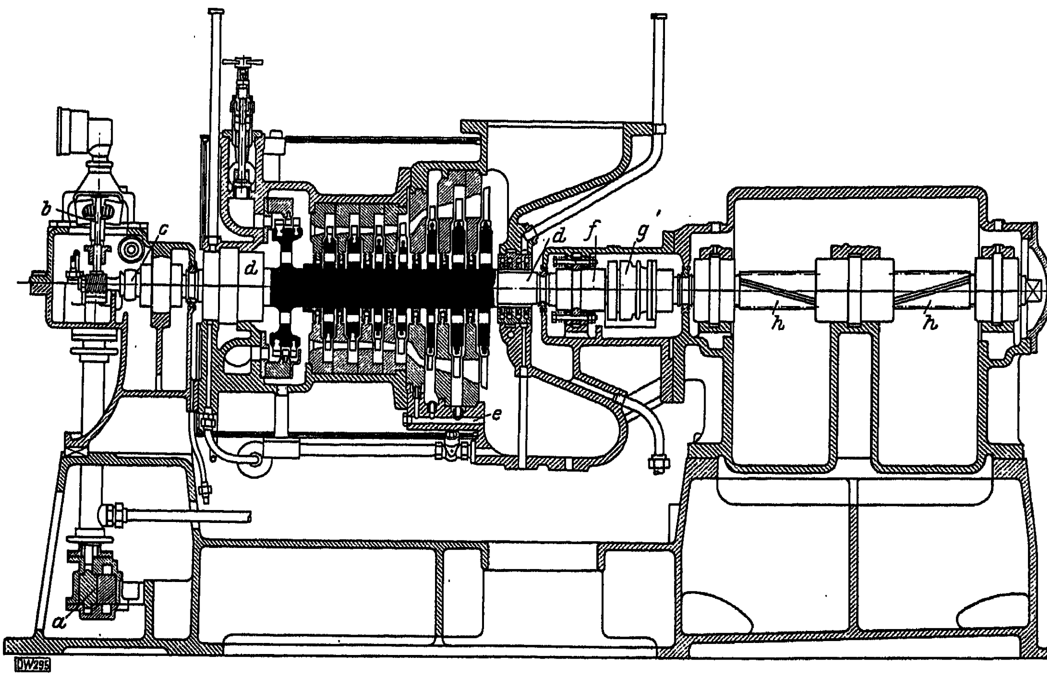


Fig. 204. *B.T.H.*, 600 kw., 8000/912 R.P.M., geared back-pressure turbine

a = Oil pump
b = Speed governor
c = Emergency governor
d = Carbon-type packing gland
e = Drain
f = Journal bearing with two thrust collars
g = Claw-type coupling with corrugated sleeve
h = Pinion with central bearing

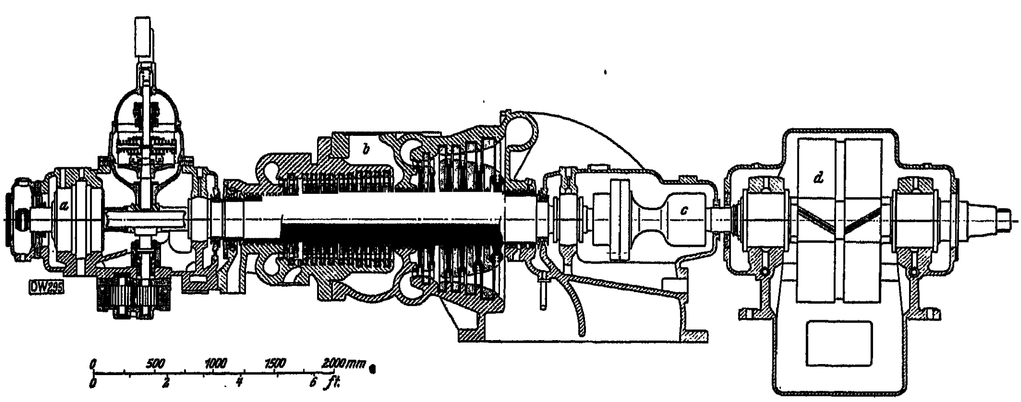


Fig. 205. *Erste Brünner*, 1000 kw., 6000/1500 R.P.M., geared extraction turbine

a = Thrust block of the pad type b = Extraction branch c = Double claw-type coupling d = Gears

When dealing with small steam quantities it is only occasionally that a deviation is made from the impulse disc type. This will happen when a small output is to be obtained from a multi-stage turbine taking steam at a high pressure. If the shaft is run below the critical speed, it will have to be of almost the same diameter as the blading and the usual form of impulse wheel is not

possible. It will then be advisable to omit the diaphragms and the special inter-stage glands. An example is the back-pressure turbine in Fig. 206. It is designed for 256 lb./sq. in. (18 kg./cm.^{2}) gauge, 700° F. (375° C.) and 28 lb./sq. in. (2 kg./cm.^{2}) gauge back-pressure.

Apart from auxiliary turbines, *B.B.C.* also build geared turbines of the characteristic reaction drum type, usually having an impulse stage for governing.

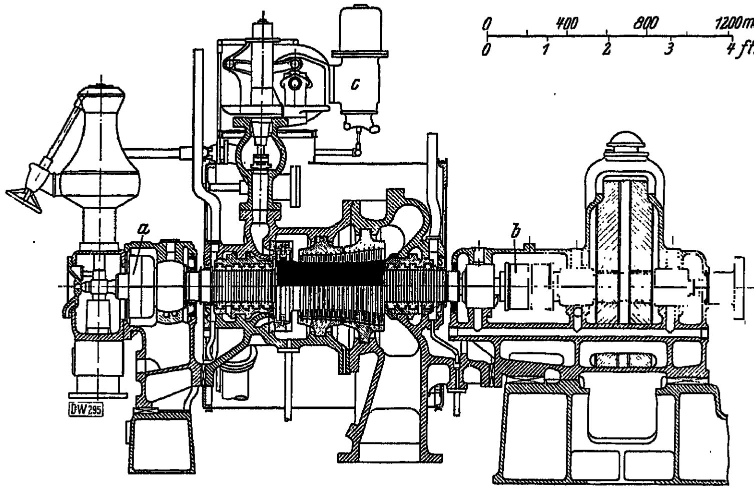


Fig. 206. *A.E.G.*, 800 kw., 7000/1000 R.P.M., geared back-pressure turbine

a = Thrust block of the pad type   b = Double toothed type coupling   c = Constant back-pressure governor

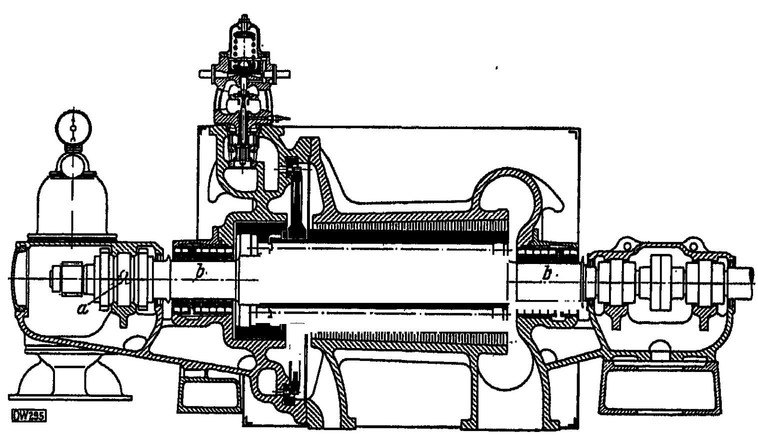


Fig. 207. *B.B.C.*, 380 kw., 5000/1500 R.P.M., geared back-pressure turbine

a = Journal bearing with double thrust bearing of the pad type   b = Carbon-type packing gland

A machine of this kind is shown in Fig. 207. The speed is reduced in gears from 5000 down to 1500 R.P.M.. The turbine has a drum with 34 stages and a large impulse wheel as first stage. It gives 380 kw. with steam at about 425 lb./sq. in. (30 kg./cm.^{2}) gauge, 700° F. (375° C.) and an exhaust pressure of 128 lb./sq. in. (9 kg./cm.^{2}) gauge.

The first cost of a small turbine set with gears is less than a direct coupled set, whilst it is somewhat higher when the output is large. The prolonged saving

in fuel should not only compensate for the higher cost but it should enable some profit to be made. By the use of gears the steam consumption of turbo-generator sets can be improved by 5 to 30%, the larger saving being obtained, naturally, for small outputs. Hence, stationary sets for large capacities will only be provided with gears when the driven machines run at exceptionally low speeds.

In Europe the usual frequency of alternating current is 50 cycles, and two-pole alternators will run at 3000 R.P.M.. This may already be less than the most economical speed for turbines of even medium capacities, especially if they are of the back-pressure type and deal with small average volumes of steam. The advantages of indirect over direct drive are particularly apparent when the turbine is coupled to a slow speed alternator or, as happens often in industrial stations, to a D.C. generator. Thus, the back-pressure turbine in Fig. 206 drives an alternator running at 1000 R.P.M. and the reduction ratio is 7 : 1. Most D.C. generators run at 1000 R.P.M. (e. g. Fig. 199) as at higher speeds difficulties in commutation are experienced.

In Germany $16^2/_3$-cycle single-phase current has been adopted for electric traction and the highest alternator speed is 1000 R.P.M.. Under present day conditions direct coupled turbines are no longer the most economical solution in power stations for electric traction, even if the sets are large, and the turbine speed should be about 3000 R.P.M.. The correct speeds can be obtained by means of gears having a 3 : 1 ratio. Fig. 208 shows the largest set of gears built for this purpose; it is also the largest set of turbine gears ever made with only one pinion. It is for 15,000 kw. and reduces the speed from 3000 down to 1000 R.P.M.. The velocity at the pitch circle is 230 ft./sec. (70 m./sec.). In order not to raise the velocity still further the pinion shaft is provided with three bearings; with only two the span would have been too great and an excessive deflection would have resulted.

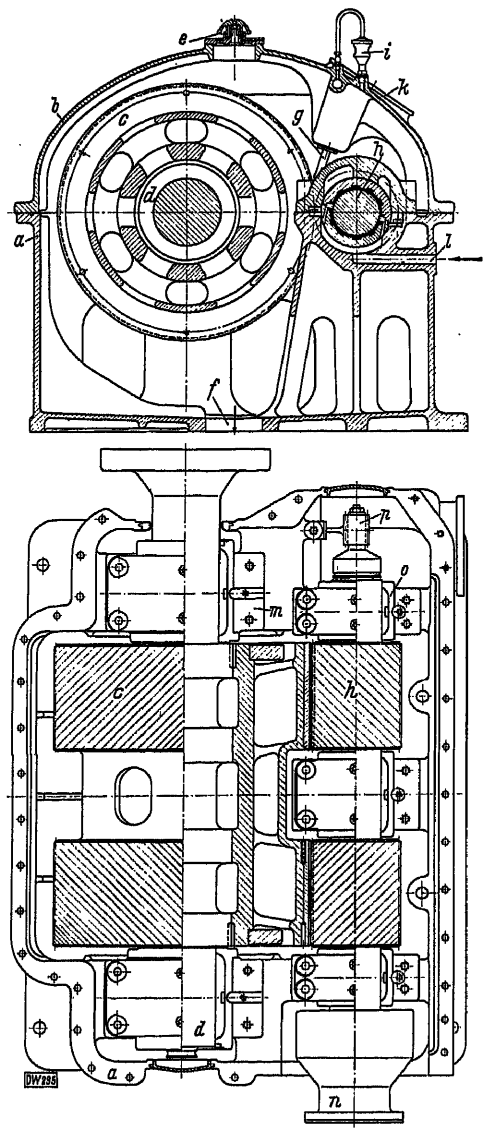


Fig. 208. *A.E.G.*, 15,000 kw. single-reduction gears for 3000/1000 R.P.M.

a = Bottom half of casing
b = Top half of casing
c = Gear wheel
d = Gear wheel shaft
e = Air vent
f = Oil outlet
g = Oil spray nozzle
h = Pinion
i = Oil pressure test fitting
k = Inspection cover
l = Oil inlet
m = Bearing of gear wheel shaft
n = Double toothed type coupling (half)
o = Bearing of pinion shaft
p = Worm drive of the gear-type oil pump

Just as steam turbines are essentially machines for large outputs, rotary pumps for gases or fluids are chiefly for large capacities. The two types of machines are well suited, therefore, for working together. Turbo-compressors require high speeds and are especially suited for direct coupling to turbines.

Rotary pumps for large outputs and high speeds cannot, however, attain the efficiencies of slow speed reciprocating pumps. Direct coupled rotary pumps usually run between 2000 to 3200 R.P.M. and are now out of date. It was only the introduction of gear drives between turbine and pump that enabled rotary pumps to outclass the reciprocating type for large outputs. Since then rotary

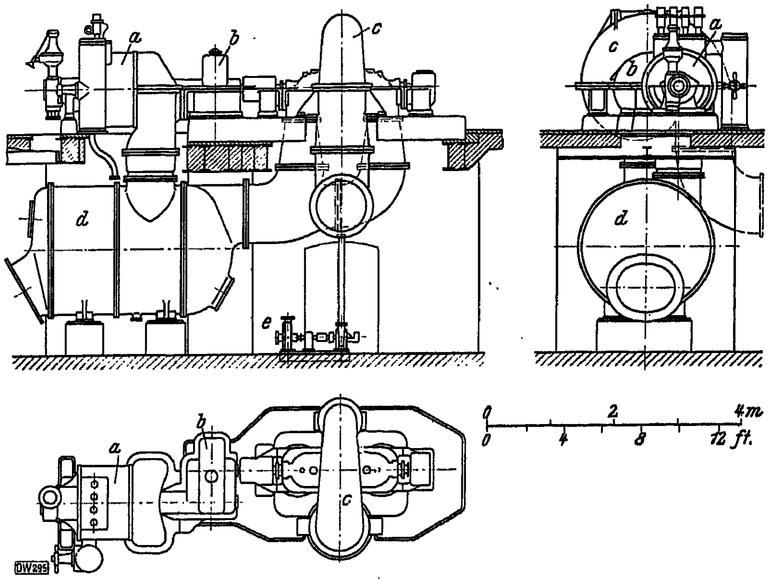


Fig. 209. *A.E.G.*, 5500/700 R.P.M. geared turbo-pump set for a water works, delivering 1,100,000 imp. gall. per hour (5000 m.^{3}/h.) against a head of 105 ft. (32 m.)

a = Turbine
b = Gears
c = Main oil pump
d = Condenser
e = Extraction pump driven by water turbine

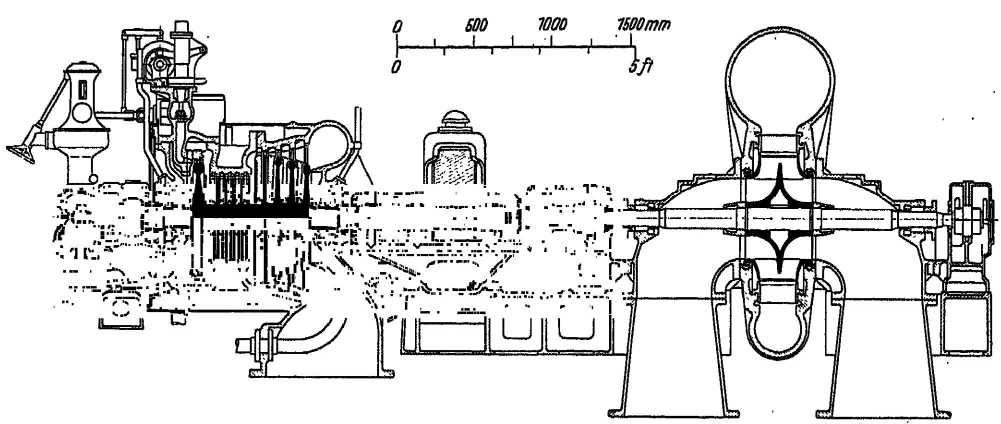


Fig. 210. *A.E.G.*, 585 B.H.P., 5500/700 R.P.M., geared high-pressure turbine driving water pump (1,100,000 imp. gall. per hour (5000 m.^{3}/h.) against 105 ft. (32 m.) head)

a = Thrust block of the pad type

pumps for indirect drive have become indispensable for water works in all large towns. The latest types can supply about 203,000 ft. lb. per. lb. of steam (62,000 kg. m./kg.), this already exceeds the performance of the best steam driven reciprocating pumps. To these general benefits must be added the even loading of the whole plant and the constant pressure in the mains. Fig. 209 gives the arrangement of a set for gear drive in a modern pumping station. 1.1 million gallons of

water per hour (5000 m.³/h.) are delivered against a head of 65 to 105 ft. (20 to 32 m.). The set consists of a high-speed turbine, gears and a centrifugal pump. All the water delivered by the main pump is drawn through the condenser and is used to condense the exhaust steam from the turbine. This arrangement is very appropriate on account of the low water temperature, which is about 50° F. (10° C.) in Europe. A separate circulating pump will not be required. A good method of driving the condensate pump in a plant of this kind is by means of a small water turbine supplied with water from the main pump. The turbine is shown in Fig. 210. The steam conditions at the nozzles are 213 lb./sq. in. (15 kg./cm.²) gauge, 660° F. (350° C.) and the vacuum is 97% at the exhaust opening. The turbine speed is 5500 R.P.M., that of the pump is 700 R.P.M..

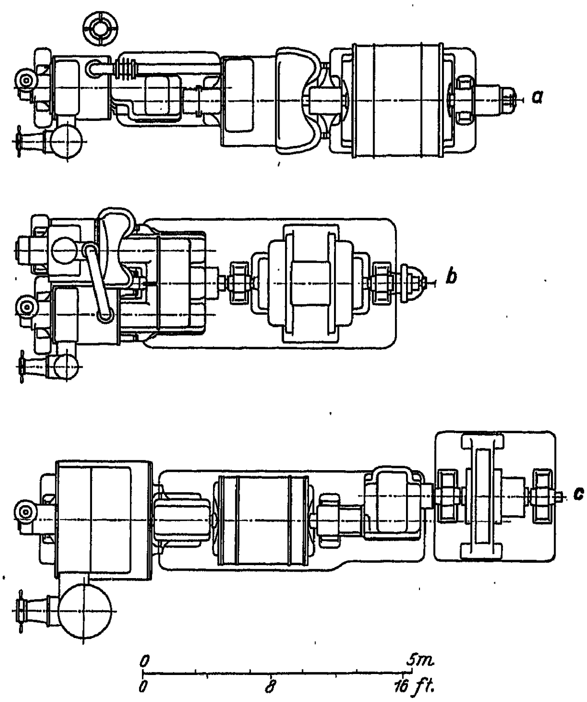


Fig. 211. Arrangement of gears in steam turbine generating sets

a = H.P. turbine — gears — L.P. turbine — alternator. Output 1600 kw. at 5000/3000 R.P.M.
b = Parallel H.P. and L.P. turbines — gears — D.C. generator. Output 1500 kw. at 5000/1000 R.P.M.
c = Turbine — alternator — gears — D.C. generator. Output 1750+750 kw. at 3000/1000 R.P.M.

In accordance with the principle of using gear drives with turbines for obtaining the best conditions of operation, it will be possible to run the separate turbines of a multi-casing set at different speeds if it should appear economical. Gears will then be used for coupling up the set (Fig. 211 *a*). This arrangement will have the further advantage that the gears will only have to transmit the output of the H.P. turbine and they will be smaller, lighter and cheaper than when they are placed between turbine and generator and have to deal with the total power. This arrangement is of particular importance for primary turbines for very high pressures. As a rule, these machines run at very high speeds and gears are used for coupling them to the following turbine with normal steam conditions and speed, or to their own generator.

Instead of having the same axis for the separate casings, the shafts may be placed parallel and the arrangement in Fig. 211 *b* will be obtained. This method can also be employed in the special case when two turbines drive a common gear wheel by means of separate pinions. A back-pressure turbine, for instance, may be coupled to one shaft and a condensing turbine to the other. The condensing turbine may be uncoupled and closed down for a time, the pinion running idle.

Other applications of gears occur when machines running at different speeds are driven by the same steam turbine. An alternator and a D.C. generator may be driven together as shown in Fig. 211 *c*. The alternator runs at 3000 R.P.M. and is directly coupled to the turbine and, as the gears are placed between the alternator and D.C. generator, they only require to be designed for the small load of the D.C. generator.

It often happens in wood-pulp works that generators and other machinery have to be driven together. The grinders run at 240 R.P.M.. The fineness of the pulp changes with even a small variation in speed and wood grinders require a very uniform drive. Steam turbines are ideal for the purpose and when gears are used they are also the most economical method of drive. The very steady running of the turbine will be increased by the momentum of the large gear wheel on the grinder shaft. It will be a good proposition to couple the turbine also to an alternator or a D.C. generator. An example may be seen in Fig. 212. The pinion drives the large wheel of the wood-pulp grinder on one side and the small wheel of the D.C. generator on the other. The turbine is connected to the pinion by means of a flexible coupling with two sets of teeth. The fixed point

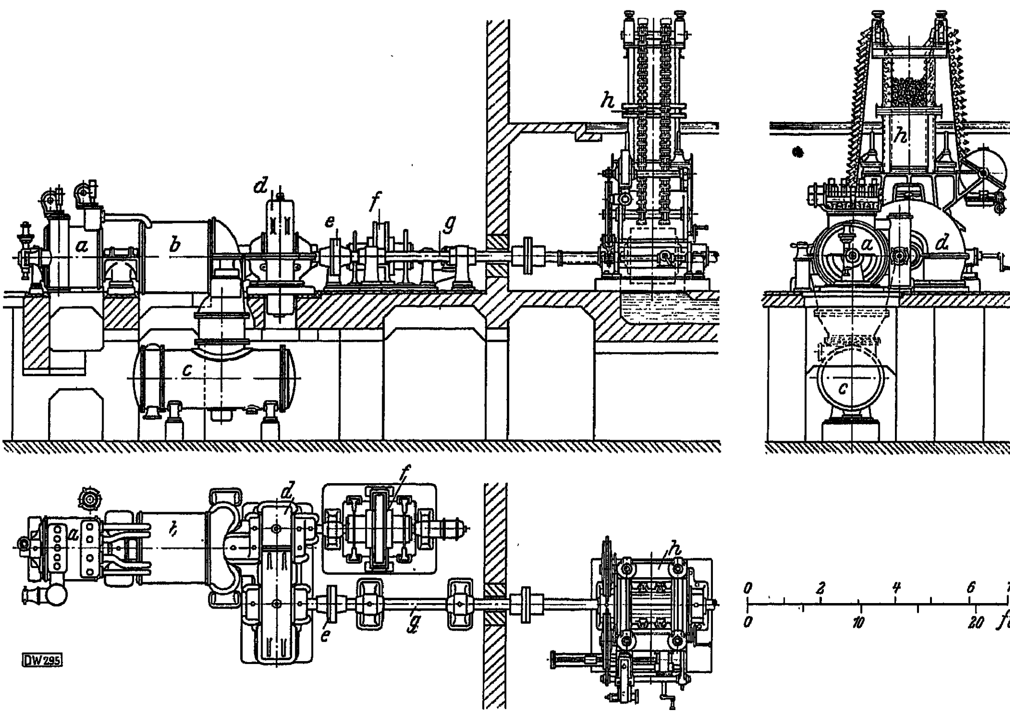


Fig. 212. *A.E.G.*, arrangement of a two-cylinder, geared extraction turbine driving wood grinder and D.C. generator. Output 2000 kw.; speeds 3000/240/750 R.P.M.

a = H.P. turbine, b = L.P. turbine, c = Condenser, d = Double gears, e = Flexible coupling, f = D.C. generator, 750 kw. at 750 R.P.M., g = Intermediate shaft, h = Wood pulp grinder, 240 R.P.M.

of the gear drive is the shaft of the large wheel which is provided with a thrust block. The generator wheel and the pinion may move in an axial direction and are free to take the best position for the teeth. The generator coupling is also of the tooth type.

In condensing plants it is useful to be able to drive several pumps with different characteristics by means of one small turbine. Gears will also be used in this case for obtaining the best speed for each machine. Fig. 213 shows a group of condenser and service pumps made by *B.B.C.* for a cargo boat. The turbine, extraction pump, oil pump and sea water pump run at the same high speed; the circulating pump is on the gear wheel shaft and runs eight times slower. On page 196 will be found the description of a pump set for a stationary condensing plant which is also driven by a steam turbine and gears.

In Fig. 214 may be seen a set with an arrangement differing from the usual. *Metro-Vick* have developed a type of turbine for outputs between 300 and 4000 kw. which does away with the cost of excavations below the engine room floor, and its auxiliaries, also, are positively driven by the main turbine. The set illustrated is for an electrical output of 1000 kw.; the turbine runs at 5000 R.P.M. and gears reduce the speed down to 1000 R.P.M. for the alternator and

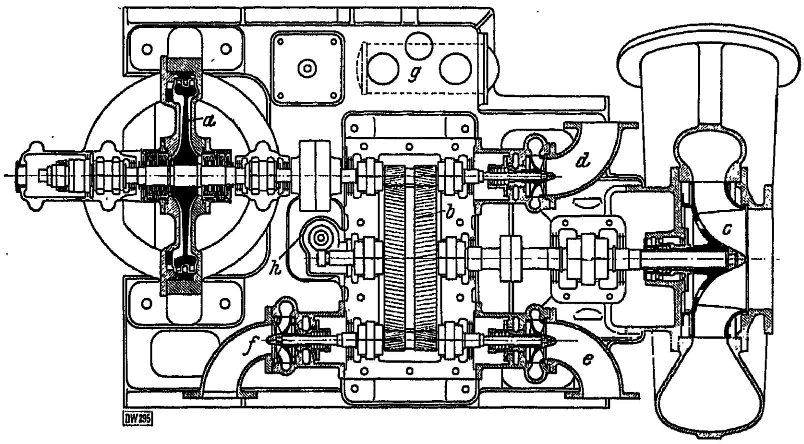


Fig. 213. *B.B.C.*, auxiliary turbo-pump set for cargo boat

a = Turbine
b = Gears
c = Circulating pump
d = Extraction pump
e = Main oil pump
f = Sea-water pump
g = Oil cooler
h = Worm drive of oil pump for the set

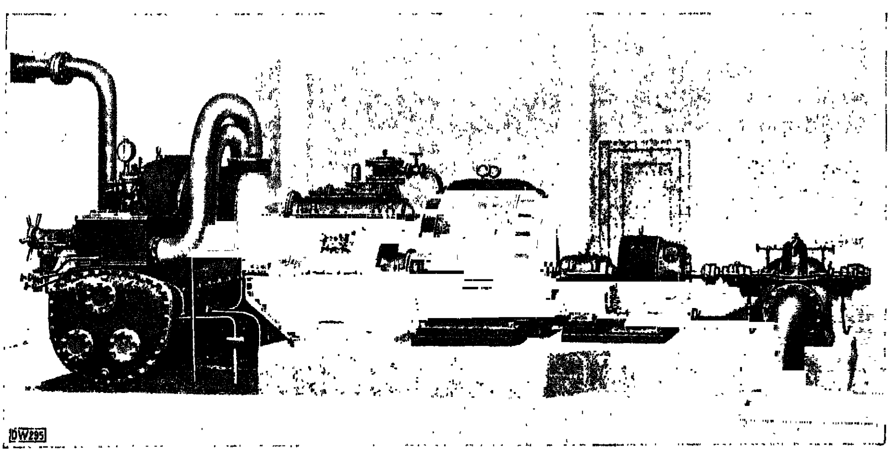


Fig. 214. *Metro Vick,* 1000 kw., 5000/1000 R.P.M., geared high-pressure turbo-alternator set

circulating pump. The gear wheel is placed underneath the pinion, thus sufficient space is obtained below the turbine for the condenser. The extraction pump is on a vertical shaft and is driven by gears at the end of the slow speed shaft of the main gears. The two-stage air ejector is mounted on the circulating water main and uses the water for condensing the ejector steam. The oil cooler is also connected to the circulating water main and the oil pump is driven by a worm on the end of the turbine shaft. The set is very accessible and easy to operate.

Gears are sometimes used for reconstructing obsolete sets of medium capacity. The original generator may be almost as efficient as an up-to-date machine, but the old direct-coupled turbine, running at a slow speed, will no longer be sufficiently economical. By using gears it will be possible to choose the speed of the new turbine without regard to that of the generator and a figure may be taken corresponding to the best efficiency. A set of this kind for 1000 kw. at 1250 R.P.M. is shown in Fig. 215 before and after reconstruction. The speed of the new turbine is 7000 R.P.M.; the original generator was kept and the gears have, therefore, a reduction ratio of 5.6 : 1. As a result of its lower steam consumption the high speed geared turbine will be amortized in a short time, especially if it is in a power station which runs day and night.

Gears have become indispensable with modern turbines for the propulsion of ships of all types and sizes. Turbines have such widely different economical speeds from propellers or paddle-wheels that the efficiency of a direct-coupled set would now be considered quite insufficient. Slow-speed turbines, directly

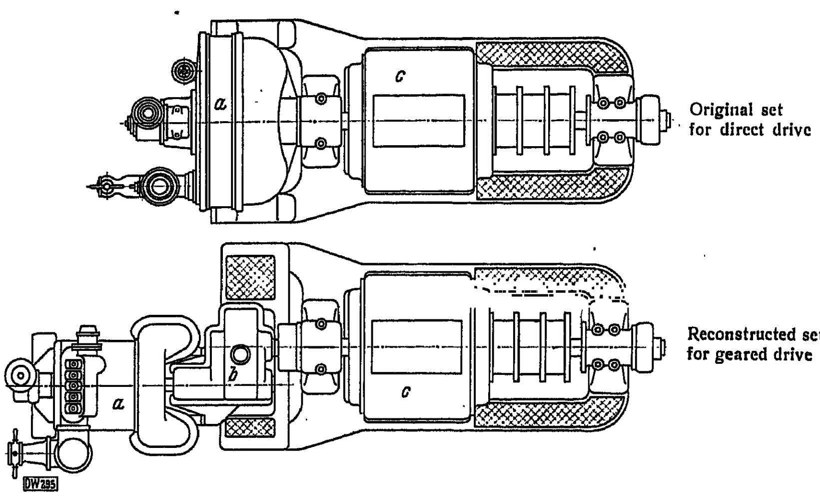


Fig. 215. *A.E.G.*, high-pressure turbo-set before and after reconstruction. Output 1000 kw., speeds 1250 and 7000/1250 R.P.M.
a = Turbine b = Gears c = Generator

coupled to the propeller shaft, were once the only machines for producing high powers and having a sufficiently small bulk and weight to be of use for ship propulsion. It was quite impossible, however, to obtain a good efficiency at slow speeds. For this reason their application was limited to fast ships, especially warships and liners. An economical solution was only obtained when gears were introduced for coupling high-speed turbines to low-speed propeller shafts, it also enabled turbine drives to become a commercial proposition when ship and propeller speeds were low. By this development the former difference, also, between slow-speed marine turbines and high-speed land turbines has almost entirely disappeared. A marine turbine may be designed even more freely than a stationary one for which a gear drive is exceptional for high loads; in a ship gears are indispensable for obtaining a good efficiency (62).

Single-reduction gears are generally used for warships and fast liners. In both these cases turbines are indisputably better than reciprocating engines on

(62) See *G. Bauer:* "Der Schiffsmaschinenbau". Vol. 2 (Munich: R. Oldenbourg 1927).

account of their smaller bulk and weight, their steady running and their better efficiency. Geared turbine plants have been built for capacities up to 200,000 H.P. on four shafts.

Turbines for slow cargo ships are usually provided with double-reduction gearing as the propeller speeds are very low and the turbines being of relatively small output are made to run at high speeds.

In spite of their small power, a few screw and paddle-wheel river steamers have been fitted with turbines and have proved most successful. The majority of applications of turbine propulsion, however, are in powerful sea going ships. As an example one of the latest single-screw fast cargo boats may be mentioned. The total output of 5400 S.H.P. is produced in two turbines in series, the same double-reduction gear being used to transmit the power from both turbines to the propeller shaft (Fig. 216). As the pinion and wheel shafts are on different levels the overall dimensions of the gear casing only depend on the size of the

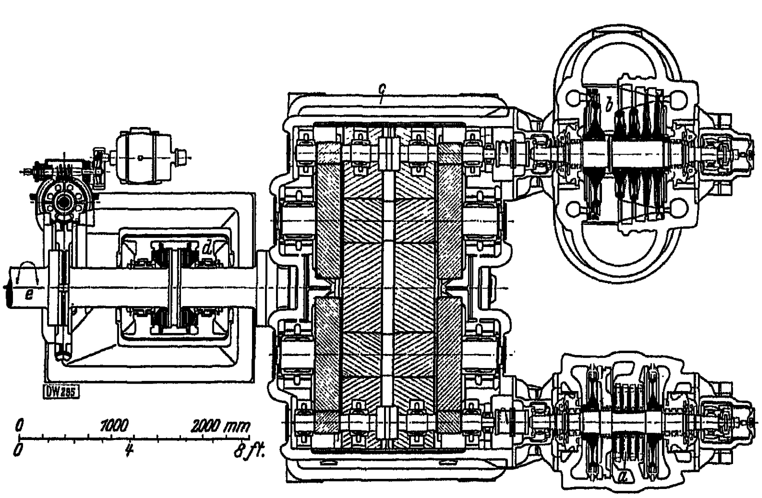


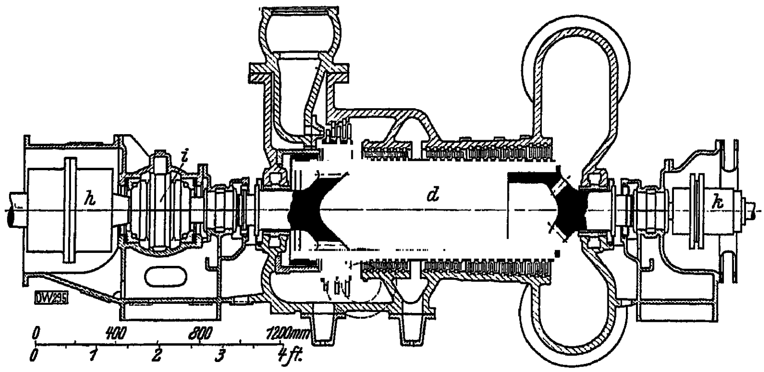
Fig. 216. *A.E.G.*, 5400 S.H.P. two-casing marine turbine with double-reduction gears. Speeds 3600/650/84 R.P.M.

a = H.P. turbine
b = L.P. turbine
c = Gears
d = Propeller shaft thrust block
e = Propeller shaft

large gear wheel. The structure of a ship always works slightly, but as the gears rest on the ship's framing within a small area, the casing will only be subject to small distortions. The two high-speed pinions are placed in the upper part of the gear casing which also supports the turbine casings. In this way it is possible to place the turbine sufficiently high to enable the condenser to be connected to an exhaust flange immediately below the L.P. turbine as in stationary sets. The flow losses to the condenser are less than with the usual arrangement in ships which is to place the condenser and turbine side by side. It may be noticed that the high speed pinion shafts are in two portions and have four bearings. The sections near the turbines are hollow and contain intermediate shafts which transmit the torque from the turbines to couplings between the portions of pinion shaft. These couplings are of the tooth type and allow for movements in an axial direction.

The *Yarrow* turbine in Figs. 217 and 218 is a typical example of a marine reaction turbine of the *Parsons* type. It drives one of the shafts of a twin screw torpedo boat. Each set gives 30,000 S.H.P. and has three casings: a H.P. turbine,

a double-flow L. P. turbine and a cruising turbine. The main set has two parallel shafts running at 3000 R.P.M. and driving the gear wheel on the propeller shaft through different pinions. The cruising turbine is for 5500 R.P.M. and is coupled to the shaft of the H.P. turbine through gears.



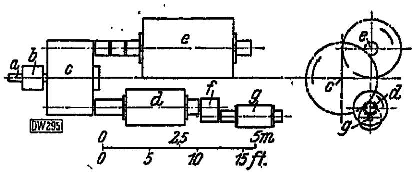


Fig. 217. *Yarrow,* arrangement and H.P. casing of a three-casing, geared marine turbo-set for 30,000 S.H.P.

a = Propeller shaft, 500 R.P.M.
b = Propeller shaft thrust block
c = Single-reduction gears
d = H.P. turbine, 3000 R.P.M.
e = L.P. turbine, 3000 R.P.M.
f = Single-reduction gears for cruising turbine
g = Cruising turbine, 5500 R.P.M.
h = Double claw-type coupling of the pinion shaft of the main gears
i = Single-collar thrust bearing with pivoted pads
k = Double claw-type coupling of the wheel shaft of the cruising turbine gears

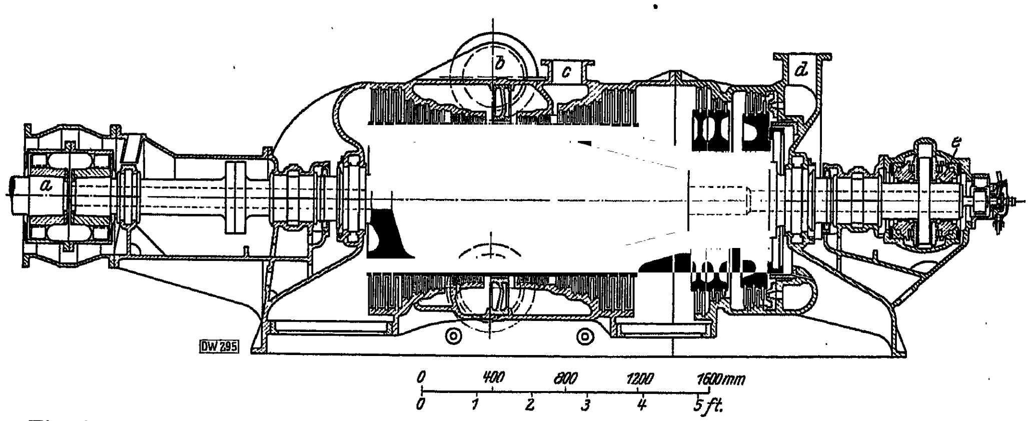
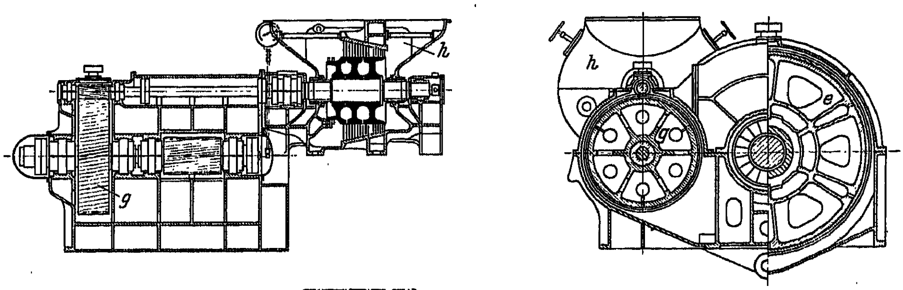


Fig. 218. *Yarrow,* double-flow L.P. casing of three-casing, geared marine turbine for 30,000 S.H.P.

a = Double claw-type coupling of the pinion shaft of the main gears
b = Steam inlet of the double-flow ahead blading
c = Branch admitting exhaust steam from auxiliary sets
d = Steam inlet of astern blading
e = Single-collar thrust bearing with pivoted pads

Attempts have been made to increase the available heat drop in locomotives by placing a condensing turbine after the reciprocating engine. Similar designs have also been tried in marine practice and have proved more successful as the space is not so limited and there is no necessity to carry the cooling water. Of the many exhaust steam turbines of this kind built for working in conjunction

with reciprocating engines, the *Bauer-Wach* type is probably the best known. The *B.B.C.* design differs from that of *Bauer-Wach* in not requiring the turbine to be disconnected when reversing (Fig. 219). The turbine has ahead and astern blading, the latter is only provided, however, to stop the turbine and enable it to start quickly in the opposite direction when manoeuvring. The turbine shown runs at 3400 R.P.M. and drives a propeller shaft at 76 R.P.M. by means of a double-reduction gear of the single helical type. The propeller shaft passes through the centre of the hollow shaft of the large gear wheel, the two being connected by means of a flexible spring coupling. The reciprocating engine alone gives 3300 I.H.P.. By providing an exhaust turbine the power was raised to the equivalent of 4250 I.H.P., or by nearly 30%, for the same initial steam conditions and steam consumption. Old ships where usually for a low power and the chief advantage of providing exhaust steam turbines after reciprocating engines is



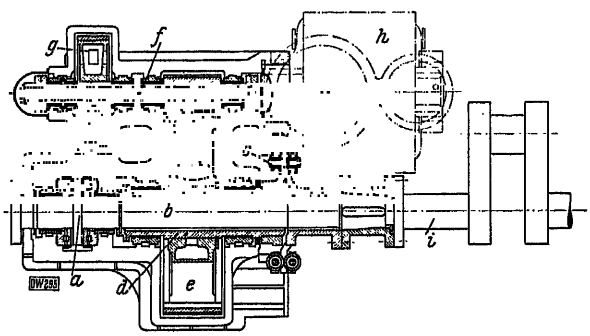


Fig. 219. *B.B.C.*, marine exhaust steam turbine coupled through double-reduction gears to the shaft of the propeller and of the main reciprocating engine. Equivalent output of turbine 950 I.H.P. at 3400/76 R.P.M.

a = Propeller shaft thrust block
b = Propeller shaft
c = Flexible coupling
d = Hollow shaft
e = Slow-speed gear wheel
f = Flexible intermediate shaft
g = Gear wheel of intermediate shaft
h = Exhaust steam turbine
i = Crank shaft of the main engine

that it enables the power and speed to be raised. As a result of the smaller coal consumption it was possible to increase also the radius of action.

Very little information has been published concerning high-speed gears and for this reason the most important points of design and construction will be briefly given.

With but few exceptions, turbine gears are of the helical type. The best spiral angle (i. e. angle between the tooth helix and the axis) is between 15° and 45° according to circumstances. The larger angles are used for high speeds. The size of the angle, however, is of secondary importance compared to the accuracy of the teeth.

As a result of the spiral angle the bearing pressure of the teeth will introduce an end thrust on the wheel and pinion shafts. If this cannot be taken up by a thrust in the other direction from the turbine or driven machinery it will be necessary to provide a collar or thrust block on the shaft. A bearing ring may also be placed close to the teeth near the circumference of the wheel and pinion according to the well-known *B.B.C.* method. The loads of single helical gears

are limited, therefore, and all large turbine gears are of the double helical type with no end thrust.

In Fig. 220 the permissible values of the bearing pressure of the teeth have been plotted in function of the pinion diameter for average conditions. These values have been established after years of practice and with the present day materials they should not be surpassed in continuous service if the gears are to have a long life. On the other hand, lower values should not be taken as the size of the gears would be unnecessarily increased and weight and price would rise. The ratio of the face width to the pinion diameter, and consequently the distance between the bearings, should not be too great as excessive bending and torsional deflections must be avoided in the pinion. The ratio of the bearing distance to the pinion diameter should not, if possible, exceed 2 : 1, whilst 2.7 : 1 may be considered the highest permissible value.

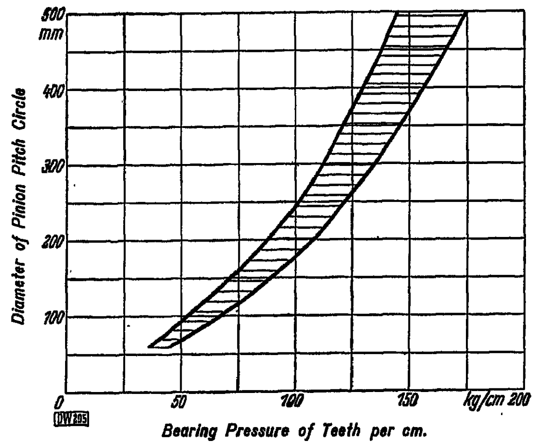


Fig. 220. Diameter of the pinion corresponding to mean bearing pressures of the teeth

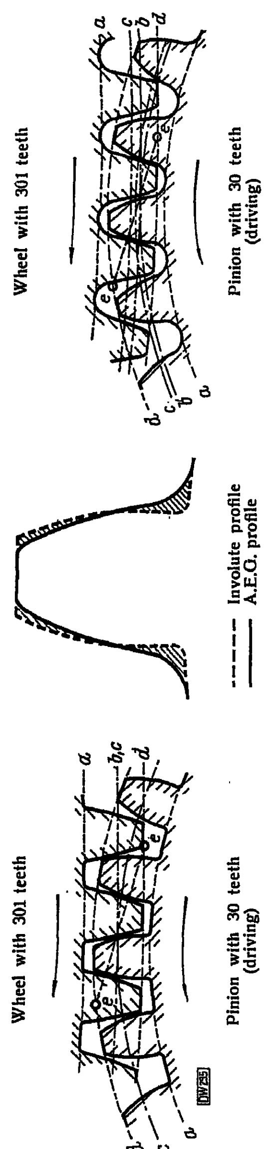


Fig. 221. Ordinary involute (left) and *A.E.G.* (right) profiles for the teeth of turbine gears

An involute profile is universally adopted for the teeth of helical gears as it possesses the following advantages: it is the simplest form to generate and allows for the greatest accuracy in manufacture; only one cutter is required for gear wheels with any number of teeth and small variations in the centre distance do not affect the tooth action. The engagement of involute teeth will always be correct provided the line of contact is sufficiently long.

The profile which has been adopted by the *A.E.G.* for gears of this kind is a modified involute. It provides for a certain sliding of the teeth in order to promote the formation of a wedge of oil in accordance with *Reynolds'* theory and in this way it avoids as far as possible any metallic contact (Fig. 221). In the case of turbine gears there is very little danger of any damage being caused by the grinding of small particles broken off the teeth as the lubrication is always copious and any foreign matter is washed away. The following rules must also be observed when designing a tooth profile. In order to obtain a strong section with a large moment of inertia, undercutting of the pinion

teeth should be avoided by an appropriate choice of the addenda and dedenda. This will considerably affect the length of the active profiles, the period of contact and the relative velocity of the surfaces. If the roots are well rounded off and the profiles are joined by a smooth, semi-circular curve the danger of notch effects will be eliminated. Towards the tip the angle between the tangent to the circle and normal to the profile can be made to increase gradually from 15° to 25°. The tips of the teeth will then be out of contact, the engagement will begin smoothly and a quiet running and good lubrication will follow. The gradual entering into contact, with the sliding increasing from zero, will prevent any friction occurring before a sufficient film of oil is formed and it will also lessen the effect of any small inaccuracies in pitch. At the ends of the toothed rings the teeth should also be chamfered and the profile slightly thinned. In this way, the ends of the teeth, which are otherwise very liable to break off, will not enter into contact.

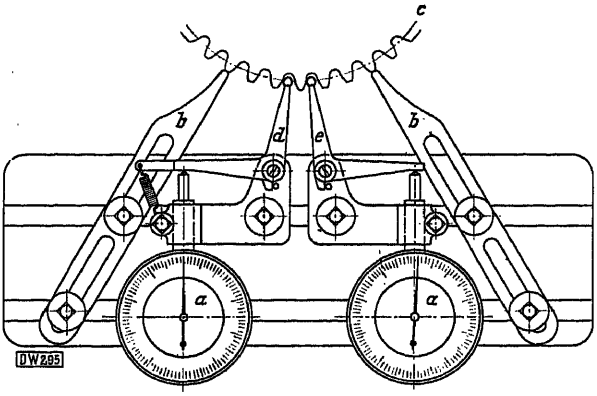


Fig. 222. *A.E.G.*, instrument for checking the accuracy of gears

a = Dial
b = Radial locating arm
c = Pitch circle of gear wheel
d = Locating lever
e = Feeler lever

The backlash should be about 0.01 to 0.02 in. (0.3 to 0.5 mm.) in order to allow for heat expansions during operation and a sufficient gap for the oil. It will be provided for when the teeth are being generated, the pitch circle of the cutter being chosen different from that of the gear. It will not be necessary then to adjust the centre distance of the wheel and pinion during erection.

Helical gears are usually generated by the hobbing process. The cutter is in the form of a spiral and it generates the teeth out of the blank in one continuous operation. Gears which are to operate silently should, naturally, be untouched by hand as it is impossible to obtain accurate and absolutely regular profiles by such methods. The so-called "running in" is also superfluous as it will never enable inaccuracies of pitch to be eliminated and the surface of the teeth should be perfectly smooth when properly manufactured.

The mean value of the errors in pitch in a high-speed gear should not exceed an arc of 5 seconds. Thus, in a wheel of 40 in. (1000 mm.) it should not be greater than about 0.0004 in. (0.01 mm.). In double helical gears the two portions are staggered so as to diminish the effect of small inaccuracies in pitch. The two sets of teeth do not start engaging at the same time and their approach and recess do not coincide.

When finished, every pinion and wheel must be verified tooth by tooth in order to find out

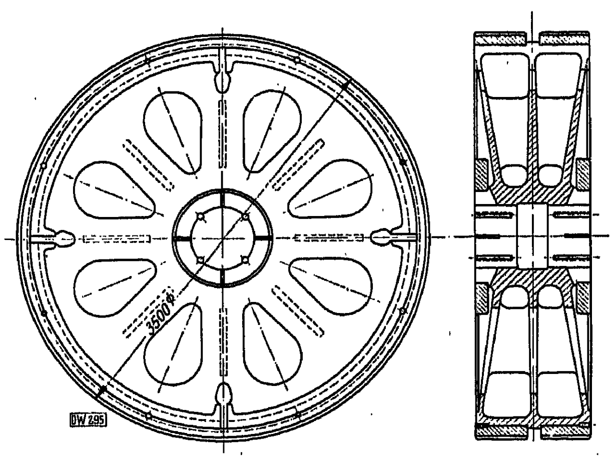


Fig. 223. *A.E.G.*, marine gear wheel (diameter 3500 mm.)

whether the permissible error has been exceeded. An instrument has been devised for this purpose and it is shown diagrammatically in Fig. 222. It is useful to keep a record of the errors in pitch of every pinion and wheel and by plotting the values on a circular chart an idea may be obtained of their distribution around the circumference. Gears with periodic errors in pitch always run noisily; they will have, therefore, a bad efficiency and will soon wear out. If, however, an accurately cut gear runs noisily, it may be said with certainty that the teeth are not engaging properly and the shafts are not correctly aligned.

The teeth of the pinion are usually cut from the solid, whilst seamless rolled rims of high-grade steel are shrunk on to the wheel. The two toothed rings of the wheel should be made together as one drum in order to obtain the same quality material.

In designing a wheel (Fig. 223) care must be taken to ensure that when the casting is cooling down or when heat expansion occurs during operation no stresses are produced which might cause distortions. After being heated the wheel is pressed on to a shaft and is held in place by two strong shrink rings. This method of fixation, relying on shrinkage only, has proved quite satisfactory, nevertheless, dowels are always provided as an additional security. For dimensioning the rims and shrink rings both experience and calculation are required. Useful values for the dimensions of rims and toothed and shrink rings of gear wheels have been plotted in Figs. 224 and 225. After the wheel has been shrunk on to the shaft, the rings are heated and put into place. They are secured by smooth and threaded dowels and the circumference is accurately turned. The wheel is then balanced for the first time

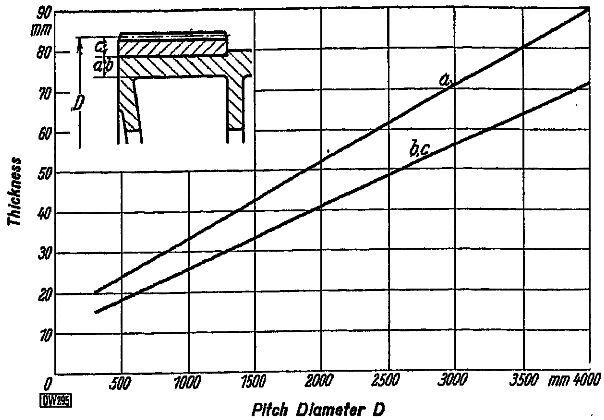


Fig. 224. Dimensions of rims and toothed rings of gear wheels

a = Thickness of cast iron rims c = Thickness of toothed rings in cast iron or cast steel
b = Thickness of cast steel rims

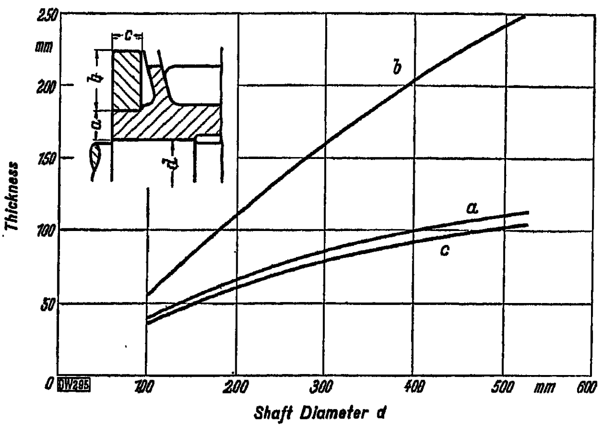


Fig. 225. Dimensions of gear wheel shrink rings

a = Thickness of hub extension c = Width of shrink ring
b = Height of shrink ring

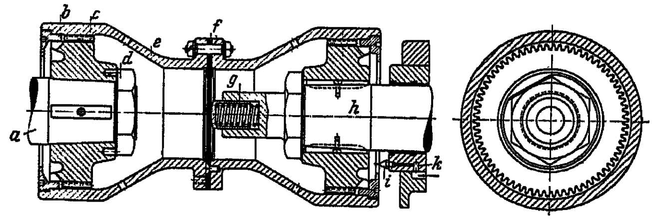


Fig. 226. *A.E.G.*, double toothed type coupling

a = Turbine shaft
b = Supporting ring
c = Toothed hub
d = Shaft nut
e = Half coupling sleeve
f = Adjusting plate
g = Helical spring with bush
h = Pinion shaft
i = Oil spray nozzle
k = Oil supply

and after the teeth have been cut it is accurately balanced again, when it is ready for erection.

For connecting the turbine shaft to high-speed double helical pinions, couplings are generally used which allow for some axial movement as the *A.E.G.* double toothed coupling, for instance (Fig. 226). In this way, the pinion is free to take the best position opposite the wheel. If this is not done the two portions will not take the same load, one half will have to transmit the whole power and a rapid wear or breakage of the teeth will follow.

The design of the casing must be adapted to the gear ratio. Small and medium sized gears may be enclosed in a casing in two parts but double-reduction gears for large powers will require a casing in several parts, particularly if the pinion and wheel shafts are on different levels in order to save space.

A common lubricating system should be provided for turbine and gears. The bearings require a thin oil and the teeth a thick oil. These two conditions may be satisfactorily met by using oils with a so-called "steep viscosity line". An oil of this kind will become very fluid in the bearings owing to the greater rise in temperature; the gears, however, will have a very good efficiency and the oil in the teeth will be only slightly warmed and will remain thick. Only mineral oils should be used in gears and they should be free from acid, tar and paraffin and should not be saponiferous. The kinds which are best suited have a viscosity figure of 200 Saybolt seconds (6° Engler) at 122° F. (50° C.). Fig. 227 shows the relation between the viscosity and temperature for such an oil. It is not advisable to use very viscous oils as the friction losses and temperatures will become excessive, especially in the bearings, there will be a great tendency to the formation of froth and the oil will have to be frequently renewed. The oil which is best suited is the one which is just viscous enough to prevent any metallic contact in bearings or gears when the internal friction is lowest. Although a less viscous oil would have a longer life, it cannot be recommended since it will be thrown off the teeth more readily, it will not be able to withstand high pressures and cannot be relied upon, therefore, for preventing the teeth wearing when in continuous service. Nozzles are usually provided for spraying the oil between the approaching teeth. The average temperatures of the oil before and after the gears are 95° F. (35° C.) and 140° F. (60° C.). The arrangement of the lubricating system should always be carefully considered.

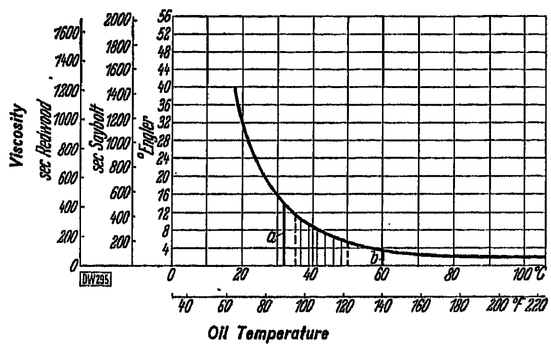


Fig. 227. Viscosity of lubricating oil for geared turbines

a = Maximum viscosity permissible
b = Minimum viscosity permissible

Normal Viscosities

3. Turbines for very high pressures

The advantages which may be gained by raising the steam pressure have already been mentioned. The present tendency is to adopt higher pressures for certain realms of steam power production, in spite of many divergent opinions. This fact is proved by the ever increasing number of plants for very high pressures in Europe and America.

New and difficult problems have arisen, consequently, in steam turbine construction. Turbines for higher pressures and temperatures built by the

different firms vary in many details but they have such a great number of points in common that they form a definite class of turbine design. This will apply wheter the machine is a condensing or an industrial turbine or whether it is directly or indirectly coupled.

The answer to the question if and when a plant for high pressures is economical—in the sense of "dollar economy"—cannot be given in this book which is exclusively devoted to the technical side of steam turbine construction. It is well known that opinions are very divided on this question.

Speaking generally, the price of fuel and the load factor will, naturally, play an important role. The way in which the decision will be affected by the first cost cannot be foretold until the difference is known between the prices of plants for normal and for high pressures. Plants using cheap coal and having small load factors should not adopt a high pressure without ample consideration. The higher the price of fuel, however, and the greater the load factor, the larger will be the probability of a high pressure being suitable. As has already been said, the correct view appears to be, with but few exceptions, that very high pressures are not yet justified in ordinary power stations but they may be used with advantage in industrial plants having extraction or back-pressure turbines.

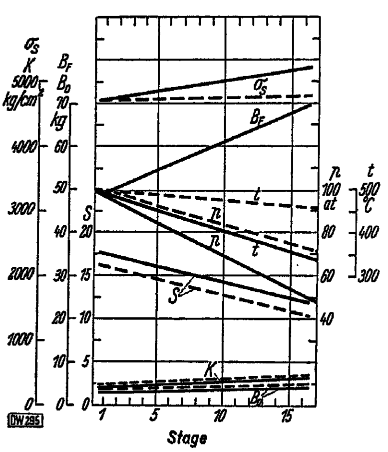


Fig. 228. Stresses in the blading of a multi-stage primary turbine for high pressures

——— at normal load
- - - - at overload
Normal load 3750 kw.
Speed 3000 R.P.M.
Live steam pressure 1400 lb./sq. in. (100 kg./cm.^{2}) gauge
Live steam temperature 930° F. (500° C.)
Back-pressure 700 lb./sq. in. (50 kg./cm.^{2}) gauge
Blade material stainless V5M steel

σ_S = Yield point, kg./cm.^{2}
B_F = Centrifugal force of blade, kg./blade
B_D = Pressure of the steam on blade, kg./blade
K = Total stress, kg./cm.^{2}
S = Safety factor = $\frac{\sigma_S}{K}$
p = Stage pressure, kg./cm.^{2} gauge
t = Stage temperature, ° C.

When designing a turbine for high pressures the engineer is always hampered by the limits of the materials and of the methods of construction which he has been accustomed to use. He certainly encounters difficulties in other fields, for instance when designing the last stage of so-called limit turbines, but those occurring in the construction of turbines for high pressures are quite different. The design, in this case, is much more influenced by the high temperatures than by mechanical stresses caused by the steam pressure. Difficulties concerning mechanical stresses do not arise in a turbine rotor in the stages for high pressures. This may be seen, for example, in the values given in Fig. 228 which are for the H.P. part of a 1400 lb./sq. in. (100 kg./cm.^{2}) multi-stage turbine for about 4000 kw. with a live steam temperature of 930° F. (500° C.). The centrifugal force of a blade will increase according to the length from about 110 to 154 lb. (50 to 70 kg.) and the pressure of the steam is never more than 22 lb. (10 kg.) per blade. Under these conditions the combined stress due to tension and bending is less than 3.2 tons per sq. in. (500 kg./cm.^{2}) even under the most unfavourable conditions (overload). The yield point of stainless steel is still about 30 tons per sq. in. (47 kg./mm.^{2}) between 750 and 930° F. (400 and 500° C.) and no blade will have a safety factor of less than

about 10 referred to the yield point. In other words, it would be possible to use much weaker blade materials in turbines for very high pressures if only they could withstand the special temperature stresses.

This example shows how greatly the design will be influenced by the high temperatures which are invariably associated with high pressures. They will considerably reduce the strength of the materials, particularly of the casing which may get distorted or displaced as a result of the large heat expansions. All casings for high pressures have to be especially designed in consequence. This may best be seen in turbines having their H.P. cylinder in forged steel. The entire casing is then machined out of a single forging. *Erste Brünner* and *Escher Wyss* employ other methods. The former provided their original 1400 lb./sq. in. (100 kg./cm.²) turbine for Witkowitz with a split cast steel

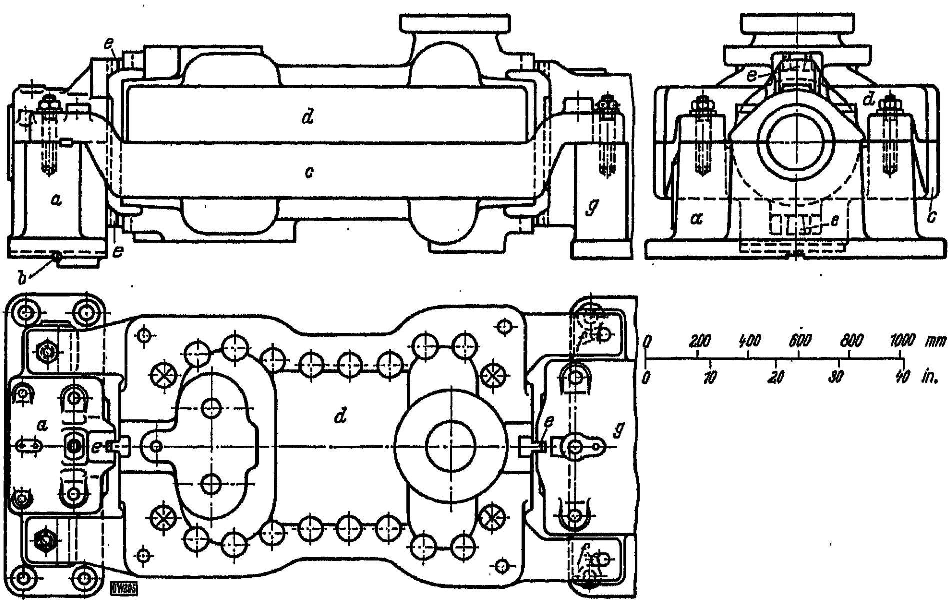


Fig. 229. *A.E.G.*, method of supporting the casing of a turbine for higher pressures on the bearing pedestal

a = Front pedestal
b = Fixed point
c = Bottom half of casing
d = Top half of casing
e = Vertical guide
f = Horizontal guide
g = Rear pedestal

casing, they surrounded it, however, with a steel sleeve in one piece. The *Benson* experimental turbine of *Escher Wyss* has a mild steel casing in one piece. Nevertheless, the use of forged steel casings has been abandoned in the majority of cases.

The most recent turbines for very high pressures in Europe and America have cast steel casings of very similar design to those for ordinary pressures. Much valuable experience has certainly been gained already in Europe where temperatures of over 750° F. (400° C.) have been used for a long time even for moderate steam pressures. The ordinary designs had to be modified, however, for obtaining a tight horizontal joint and for maintaining the correct alignment of casing and rotor. This may be seen on the H.P. casing of a 1400 lb./sq. in. (100 kg./cm.²) turbine for an initial temperature between 840 and 890° F. (450 and 475° C.) and a back-pressure of 710 to 1070 lb./sq. in. (50 to 75 kg./cm.²) (Fig. 229). The flanges are extremely thick and the bolts at the horizontal joint are very close together. Not only were the bolts made of the best material

but a special method, which has already been mentioned, was used with complete success for tightening them. Before the nuts were screwed down, the bolts were heated and on cooling down they contracted and applied additional pressure. The feet of the casing have been placed level with the horizontal joint and as near as possible to the bearing centres. Thus, distortions due to the heat are avoided, also a correct alignment of the rotor is maintained. The casing rests, therefore, on two feet on either side of the bearing, it is also provided with vertical guides above and below the bearing. Hence, it will be free to expand upwards or downwards from the horizontal joint or, in other words, in practically any radial direction.

With regard to other parts in turbines for higher pressures, it has been found that for the present day conditions, or for pressures not exceeding about 2800 lb./sq. in. (200 kg./cm.^{2}), there is no necessity for any new details differing essentially from those employed in turbines for ordinary pressures. The rotors have usually many stages of small diameter and are often cut from the solid. Owing to the small volume of the steam, impulse blading is more frequently used than reaction blading. It is even possible to retain the ordinary designs of glands, the type in three concentric parts employed by the *Amer. G.E.C.* being no exception to this rule since it is also used for moderate pressures and temperatures. It is only for pipes, valves and other fittings in the live steam main that it has been necessary to devise new methods, only two of which will be mentioned here. They are both of particular interest as they are practical applications of welding, an operation which has recently been acquiring popularity as a method of manufacture. The seats of valves for live steam used to be made, generally, of pure nickel and they were pressed into the cage or valve. They are manufactured to-day out of the much more resistant V2A steel, a nickel-chromium steel belonging to the austenitic series, and they may satisfactorily be welded to the valve. Cast steel and nickel-chromium steel may now be perfectly welded together. The welding of pipe joints for live steam is of even greater importance. When it is difficult to obtain tight joints with flanges, an obvious thing to do is to weld the pipes together. The applications of this method have proved quite successful up to the present, but pipes which have been joined in this way can only be separated by an oxy-acetylene blow lamp. For this reason, it may seem better to make joints which can be broken and to continue using flanges, these should, however, be welded to the pipe.

A welded joint must be filled up with fluid metal, hence it will be somewhat similar to a casting; it is impossible, therefore, that it will be of equal quality to the original material of the rolled pipes. In addition to this disadvantage, much experience and care is needed in order to obtain a joint free from slag and inclusions of gas. A swelling is always provided at the joint and the wall is made slightly thicker than for the piping so as to reduce the stresses. The welding may be further improved if the joint is lightly hammered whilst it is white hot, the operation should be stopped at red heat, and should on no account be continued down to blue heat. Finally, the mechanical properties of a weld may also be improved by tempering.

When enclosures of slag or gases occur, it is usually near the surface, for this reason it is advisable to avoid any bending in welded joints as it is well known that the outer fibres will then be the most highly stressed. In order to obtain good welds, reliable and experienced craftsmen are, naturally, required, and samples should frequently be taken in order to test the uniformity of the work. It has been found by numerous tests, however, that it is possible to produce perfect joints without difficulty and no decrease in strength is apparent either in the weld itself or in the neighbouring metal (Fig. 230). Apart from welding, joints in high-pressure steam pipes may be made successfully with

Magnified 3 times

Magnified 100 times

Fig. 230. Microphotographs of welded joint and structure of the transition zone of a live steam pipe of S.M. steel

flanges which are either screwed, rolled or riveted to the pipes. Long experience only will disclose which method is the best.

It is not yet possible to give any universal rules for the design of turbines for high pressures. However, the general trend of developments may be shown and they have already been described (63). Many of the plants which have been built up to the present must only be considered as experiments. The design of the various parts differs according to the firm, but the same aim is apparent in all instances; it is to make casings which will withstand high temperatures and pressures, without their rigidity being affected, and may freely expand in operation. This may best be seen in designs of turbines already executed. When selecting the following examples preference was given to commercial machines rather than to those built especially as experiments.

If the history of the adoption of high pressures in Europe is considered, it will be found that *de Laval* was the pioneer. As early as 1897, he made a boiler and turbine for no less than 2800 lb./sq. in. (200 kg./cm.^{2}) with 750° F. (400° C.). They supplied the power to the Stockholm Exhibition. *De Laval* has also made some more recent turbines for high pressures (Fig. 231), they are small back-pressure turbines for about 500 kw. with steam at 1400 lb./sq. in. (100 kg./cm.^{2}), 750° F. (400° C.). Naturally, for this range of output, a turbine for high pressures will only be justified as a back-pressure machine because it may then work at a poor efficiency with but little effect on the overall efficiency of the plant. The entire heat drop is employed in two single-row impulse stages, overhung at the end of the shaft. The mean diameter of the blading is only 11.6 in. (295 mm.) which enables a sufficient blade height to be obtained with full admission, notwithstanding the small volume of the steam. The turbine speed is 15,000 R.P.M..

The first applications in Europe of very high pressures on a large scale were made at the Langerbrugge Power Station in Belgium. Before the pressure was raised, all the turbines in the station were for 285 lb./sq. in. (20 kg./cm.^{2}) gauge. The primary turbine built by *B.B.C.* works between the high-pressure boilers for 710 lb./sq. in. (50 kg./cm.^{2}) gauge, 830° F. (440° C.) and the old

(63) See *G. Forner:* "Die Dampfturbine für Betrieb mit Höchstdruckdampf", Brennstoff- und Wärmewirtschaft 7 (1925) p. 41.

steam main. It is in two casings, each with two impulse stages, and the total output is 1675 kw. at 8000 R.P.M.. A cross-compound arrangement was adopted and the two turbine shafts drive one 1500 R.P.M. alternator through the same set of gears. The difficulty of providing H.P. glands has been avoided here also and the wheels are overhung at the end of the pinion shafts. As a temperature of 830° F. (440° C.) was unusually high at the time when the set was built (1924), the casings were made of high quality electro-furnace cast steel. They are overhung and can expand freely, radial guides being provided. The steam connections between turbine and valve chest and between the two casings are effected by several small pipes which are very flexible and no distortions can be caused by their thrusts.

Overhung primary turbines are entirely satisfactory for relatively small outputs. For larger powers, however, it is impossible to avoid the ordinary

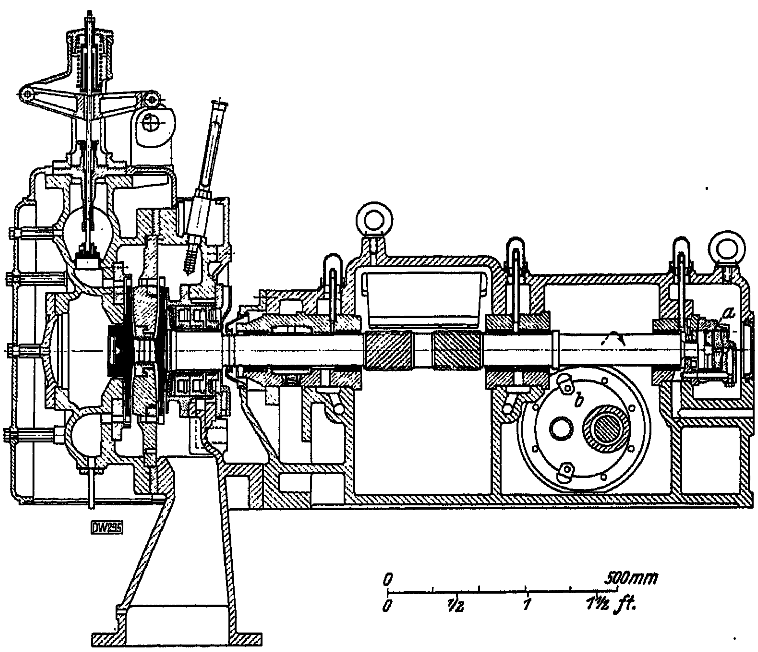


Fig. 231. *De Laval,* 500 kw., 15,000/3000 R.P.M., geared back-pressure turbine for high pressures

a = Thrust block of the pad type   *b* = Governor and oil pump drive

type of rotor with two bearings. There is a great similarity between turbines of this design; it is even greater than between types for normal pressures because the L.P. part is separate and need not be considered at present. Impulse turbines of all makes have solid rotors, as the 30-stage rotor in the first casing of the 18,000 kw. *Erste Brünner* set for Witkowitz, or as the first primary turbine for the Edgar Station at Boston made by the *Amer. G.E.C.* which has 20 stages and is equally well known. Fig. 232 shows a larger and more recent turbine made by the *Amer. G.E.C.* for Boston also. This turbine gives 10,000 kw. at 3600 R.P.M. and takes steam at 1200 lb./sq. in. (84 kg./cm.^{2}) gauge, 700° F. (370° C.). It has 16 impulse stages which are solid with the shaft. The casing is in mild steel. The first eight diaphragms are held in grooves in the casing whilst the last ones are carried on a separate sleeve. Triple-flow glands are provided at both ends of the turbine. The steam strainer is placed in the live steam belt immediately before the first stage.

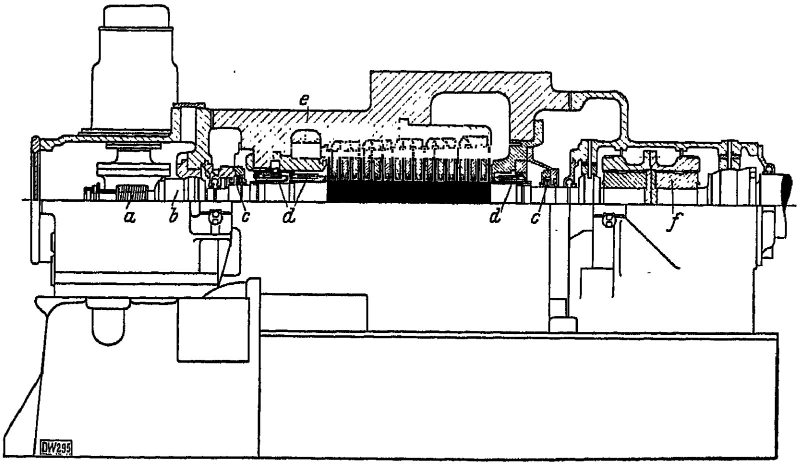


Fig. 232. *Amer. G.E.C.*, 10,000 kw., 3600 R.P.M. primary turbine for high pressures

a = Governor drive
b = Thrust block of the multi-collar type
c = Water-sealed gland
d = Three-flow labyrinth-type packing gland
e = Steam strainer
f = Double claw-type coupling

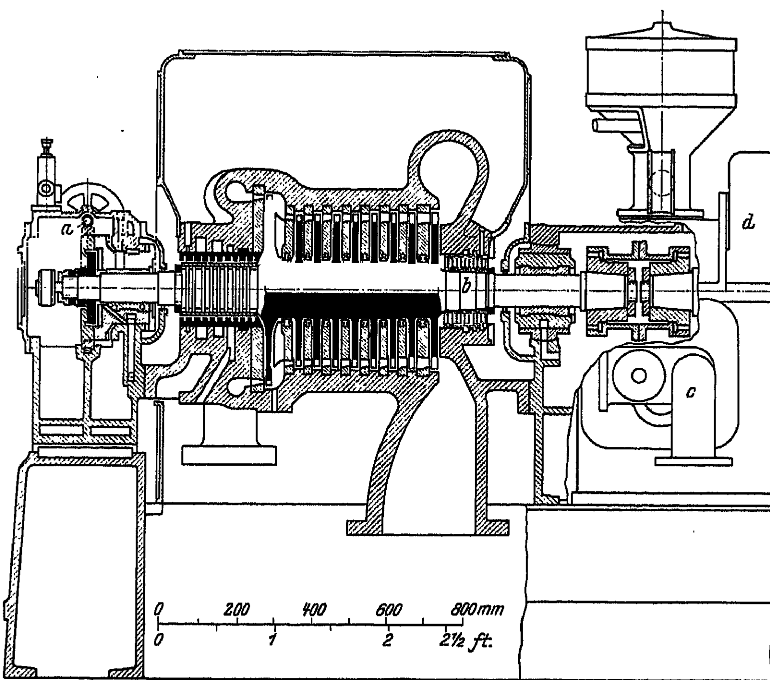


Fig. 233. *English Electric*, 2500 kw., 6000/1000 R.P.M. primary turbine for high pressures

a = Rotor adjusting gear
b = *Ljungström*-type packing gland
c = Oil pump
d = Gears

The nine-stage turbine for 1000 lb./sq. in. (70 kg./cm.^{2}) built by the *English Electric* for Bradford (Fig. 233) has a very similar appearance to the standard designs of the firm. It gives 2500 kw. at 6000 R.P.M. and drives a 1000 R.P.M. alternator through gears. The initial temperature of the steam is 770° F. (410° C.) and the back-pressure 210 lb./sq. in. (14.5 kg./cm.^{2}) gauge.

The diameters of the stages are 19 and 15 in. (483 and 380 mm.). The blades increase gradually in length from 0.45 to 1.08 in. (11.5 to 27.5 mm.). The turbine is designed to exhaust into the old steam main. Before building this commercial turbine, the *English Electric* had already made a small single-stage experimental turbine for 360 kw. at 25,000 R.P.M. supplied with steam from a *Benson* boiler (64). *S.S.W.* made some similar experiments in conjunction with *Escher-Wyss* who supplied the turbines.

The definition of steam turbines for high pressures may logically be extended to include turbines which have been made as experiments for very high steam temperatures, the pressure, however, being only moderate. The most outstanding example of a turbine of this type is the 10,000 kw. *B.T.H.* machine for the Delray Station at Detroit. This turbine expands steam from about 370 lb./sq. in. (26 kg./cm.^{2}) gauge at 1000° F. (540° C.) down to 0.5 lb./sq. in. (0.035 kg./cm.^{2}) absolute without any reheating. It has two cylinders and the steam flow is about 90,000 lb./h. (40,000 kg./h.). In order to reduce the radiation losses of the hot steam mains, separate oil-fired superheaters are placed close to the turbine. The plant must be considered chiefly as an experiment and for this reason the principal aim was to obtain a reliable operation rather than a high efficiency. Nevertheless, it is expected that in this latter respect the values will be equal or even superior to those of a high-pressure plant.

For obtaining data on applications of high pressures it is not sufficient to employ only experimental plants but it is necessary to build plants of similar sizes to those required in practice. It is with this object in view that the first power stations

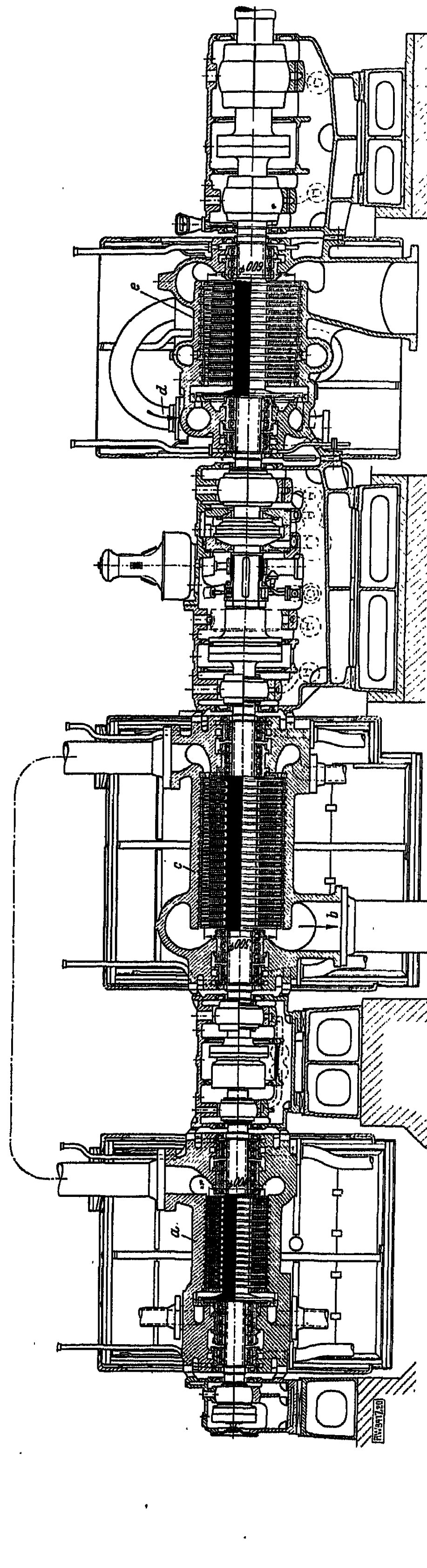


Fig. 234. *A.E.G.*, 12,000 kw., 3000 R.P.M., three-casing, extraction back-pressure turbine for high pressures (dimensions in mm.)

a = H.P. casing *b* = To reheater *c* = I.P. casing *d* = From reheater *e* = L.P. casing

(64) See *A. H. Law* and *J. P. Chittenden:* "Higher steam pressures and their application to the steam turbine", Engineering 124 (1927) pp. 610, 728 and 763.

for very high pressures, such as Boston, Langerbrugge, or Mannheim, were built. It was for this reason also that the first and largest industrial high-pressure plant was installed in one of the foremost lignite works in Germany. This firm had previously conducted experiments on new applications of steam plants with great care and courage. The plant (65) consists of two three-cylinder extraction back-pressure turbines. Each turbine (Fig. 234) requires at its normal load of 12,000 kw. about 220,000 lb./h. (100,000 kg./h.) of live steam at 1400 lb./sq. in. (100 kg./cm.^{2}) gauge and 840° F. (450° C.). As maximum values 1570 lb./sq. in. (110 kg./cm.^{2}) gauge, 880° F. (470° C.) are permitted. After the second casing about 90,000 lb./h. (40,000 kg./h.) are extracted at 178 lb./sq. in. (12.5 kg./cm.^{2}) gauge; the remainder of the steam is taken to a reheater, which obtains its heat by condensing live steam, and expands afterwards down to 35.6 lb./sq. in. (2.5 kg./cm.^{2}) gauge in the third casing. The three casings have a common shaft running at 3000 R.P.M.. The steam quantities are so large that sufficiently long blades are obtained even in the H.P. part and flow losses are small. All the casings are split horizontally. The coupling between the H.P. and I.P. rotors is flexible and the fixed point of the H.P. casing is the first bearing pedestal. A thrust bearing is also provided here. The I.P. and L.P. rotors are rigidly coupled and are arranged for flows in opposite directions in order to balance the thrust. Any unbalanced thrust is taken up by a second thrust block in front of the L.P. turbine. The corresponding bearing pedestal houses the emergency governor and the drive for the main governor. The L.P. turbine is bolted in the usual way to the pedestals with keys as guides, but the H.P. and I.P. casings are supported on feet resting on the bearing pedestal level with the shaft, according to the method already mentioned.

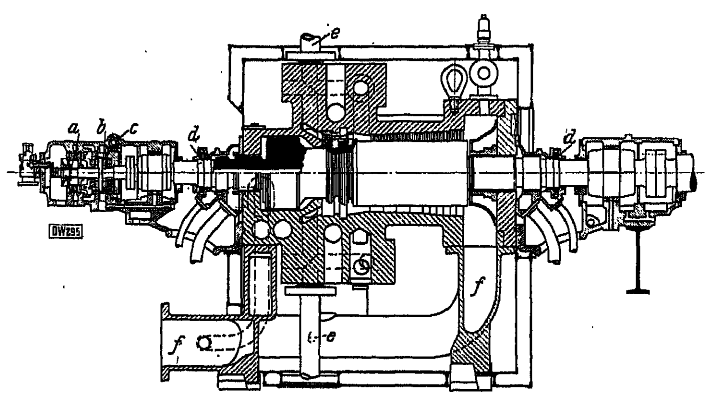


Fig. 235. *Westinghouse,* 10,000 kw., 3600 R.P.M. primary turbine for high pressures

a = Oil impeller for governing
b = Thrust bearing of the pad type
c = Rotor adjusting gear
d = Water-sealed gland
e = Live steam pipe
f = Exhaust branch

Reaction turbines for high pressures have already been built by *Westinghouse* and *B.B.C.*. Both firms use a two-row Curtis wheel as first stage. Fig. 235 illustrates a 10,000 kw. *Westinghouse* turbine. The working conditions are 1200 lb./sq. in. (84 kg./cm.^{2}), about 700° F. (365° C.) and an exhaust pressure of about 310 lb./sq. in. (22 kg./cm.^{2}) gauge. The turbine runs at 3600 R.P.M.. The casing is machined out of a steel forging. The rotor is a solid drum and the reaction blading is end tightened. The H.P. gland and the balance piston are also sealed in an axial direction. A gear is provided for adjusting the position of the rotor. It moves the thrust block and the entire rotor is then shifted in an axial direction if necessary when starting up or closing down (i. e. when the casing and rotor are heating up or cooling down) and the correct blade and packing clearances may be obtained.

Finally, Fig. 236 shows the primary turbine made by *B.B.C.* for the Mannheim Power Station. It is for 7000 kw. at 3000 R.P.M. and expands steam from

(65) Refer to foot-note 15 on page 16.

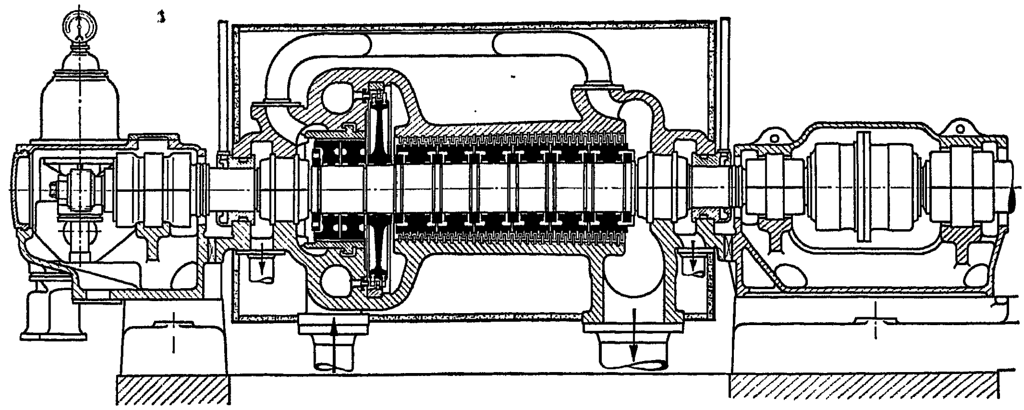


Fig. 236. *B.B.C.*, 7000 kw., 3000 R.P.M. primary turbine for high pressures

1400 lb./sq. in. (100 kg./cm.²), 800° F. (430° C.) down to 257 lb./sq. in. (18 kg./cm.²) gauge. The exhaust steam passes through a surface reheater taking heat from live steam. A different design of rotor has been adopted to that of the *Westinghouse* machine, the stages being divided into groups and mounted on separate discs on the shaft. This was done in order to obtain a quick and even heating of the rotor when starting up. The ordinary *B.B.C.* method has been employed for mounting the discs on the shaft (refer Fig. 84). The casing is in cast steel. As in most of the turbines which have been mentioned, all valves are placed at the side of the turbine in order to simplify the form of the casing as much as possible.

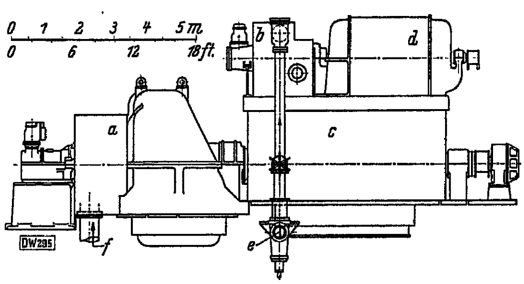


Fig. 237. *Amer. G.E.C.*, 50,000 kw., vertical-compound turbo-alternator set Speed of H.P. Turbine 3600 R.P.M., of L.P. Turbine 1800 R.P.M.

a = L.P. turbine
b = H.P. turbine
c = Alternator, 37,500 kw.
d = Alternator, 12,500 kw.
e = Live steam pipe
f = Pipe from reheater

The great difference in size between the small H.P. part and the large L.P. part of turbines for high pressures has led to an original arrangement in America. The design was also influenced by the lack of space in stations in the centre of large towns. In such cases the *Amer. G.E.C.* place the primary turbine and its alternator on top of the L.P. alternator (Fig. 237). The total capacity of the set illustrated is 50,000 kw.. The primary turbine is for 1200 lb./sq. in. (84 kg./cm.²) and runs at 3600 R.P.M.; the speed of the L.P. turbine is 1800 R.P.M..

The largest unit of this type is now being constructed for the *Ford* Works (Fig. 238). It will give no less than 110,000 kw.. This machine differs in many details from the previous example mainly because both parts run at 1800 R.P.M.. Each casing is for 55,000 kw.. Consequently, the H.P. part is of larger dimensions than previously and the H.P. turbine is placed on top of the L.P. turbine. The initial pressure is also 1200 lb./sq. in. (84 kg./cm.²) and the temperature 725° F. (385° C.). The set will probably prove entirely satisfactory for the special conditions for which it was designed.

Few attempts have been made to apply higher steam pressures to other than stationary plants. The highest pressure which has yet been used in ships is not more than about 500 lb./sq. in. (35 kg./cm.²). Experimental locomotives for

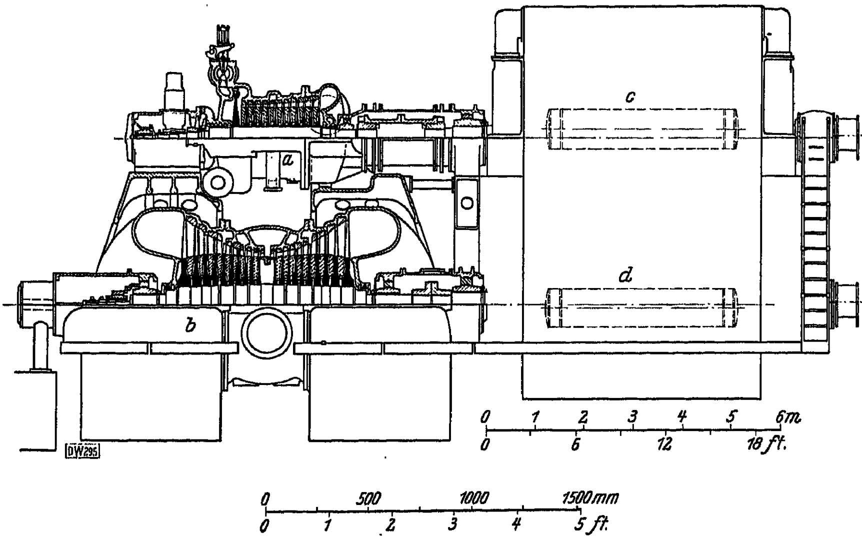


Fig. 238. *Amer. G.E.C.*, 110,000 kw., 1800 R.P.M., vertical-compound turbo-alternator set

a = H.P. turbine c = Alternator, 55,000 kw.
b = L.P. turbine d = Alternator, 55,000 kw.

high pressures up to 1700 lb./sp. in. (120 kg./cm.²) having reciprocating engines or turbines have been designed or are under construction. Operating or test results are not yet available.

Owing to the lack of sufficient results of steam consumption and efficiency tests of high-pressure plants, it is unfortunately not possible to give any definite opinion even on stationary plants. However, the possibility of building and operating stations of this kind with sufficient reliability has certainly been proved by the Edgar Station in Boston, where turbines have been working since 1924 with 1200 lb./sq. in. (84 kg./cm.²), and by the Langerbrugge Power Station. Full reports on the operation of these plants have frequently been published, in the first-named case especially (66). Meanwhile a number of other high-pressure stations has been put into service and it may be hoped that in the near future practical results will give their verdict on the question of high pressures which so far has solely been discussed in general theoretical terms. It has only been possible to mention a few high-pressure plants; if a general survey is required reference may be made to Table II (67) giving the most important data on all known high-pressure turbo-plants and references to publications (see the following pages).

(66) See the various articles by *J. E. Moultrop* published in Power since 1924.
(67) Compiled with the help of a list given by *G. A. Orrok*, Power 67 (1928) p. 340.

Table II. Turbines for high pressures

Station	p_1 (Turb.) lb./sq. in. (kg./cm.²) gauge	t_1 (Turb.) °F. (°C.)	p_2 lb./sq. in. (kg./cm.²) gauge	Reheating at lb./sq. in. (kg./cm.²) gauge	Regenerative Feed-heating stages	Unit Capacity kw.	Speed R. P. M.
Delray No. 3, Detroit, U. S. A.	365 (26)	1000 (538)	Cond.	—		10,000	
Arnau Paper Mills, Czecho-Slovakia	710 (50)	750 (400)	Double extraction cond.	—	—	1850	7500
Langerbrugge, Belgium	710 (50)	825 (440)	300 (21)	—	—	1675	8000
„	710 (50)	825 (440)	300 (21)	—	—	8000	
„	710 (50)	825 (440)	Cond.	—	4	25,000	3000
Gothenburg, Sweden	810 (57)	842 (450)					6000
Solvay, Syracuse, U. S. A.	840 (59)	735 (390)	164 (11.5)	—	—	5000	3600
Valley Road, Bradford, England	1000 (70)	775 (410)	207 (14.5)	—	—	2500	6000
Edgar, Boston, U. S. A.	1200 (84)	700 (370)	360 (25.3)	—	—	3150	3600
„	1200 (84)	700 (370)	360 (25.3)	—	—	10,000	3600
„	1200 (84)					12,500	3600
Calumet, Chicago, U. S. A.	1200 (84)	750 (400)	313 (22)	313 (22)	—	5000	3000
Lakeside, Milwaukee, U. S. A.	1200 (84)	750 (400)	300 (21)	300 (21)	—	7000	3600
„	1200 (84)	750 (400)	300 (21)	300 (21)	—	7700	3600
North East, Kansas City, U. S. A.	1200 (84)	725 (385)	313 (22)	313 (22)	—	10,000	3600
Deepwater, N. Y., U. S. A.	1200 (84)	750 (400)	200 (14)	—	—	12,500	3600
„	1200 (84)	750 (400)	Cond.	400 (28)		53,000	3600/1800
San Francisco, Cal., U. S. A.	1200 (84)	750 (400)	Cond.			50,000	3600/1800
Fordson, Michigan, U. S. A.	1200 (84)	725 (385)	Cond.			110,000	1800
Holland, N. Y., U. S. A.	1250 (88)	750 (400)	Cond.	416 (29)	4	55,750	3600/1800
South Amboy, N. Y., U. S. A.	1250 (88)	750 (400)	Cond.			25,000	3600/1800
Stockholm, Sweden	1400 (100)						
Fors Bruk, Sweden	1400 (100)	825 (440)	85 (6)	—	—	480	15,000
Mannheim, Germany	1400 (100)	800 (430)	256 (18)	256 (18)	—	4800	3000
„	1400 (100)	800 (430)	256 (18)	256 (18)	—	7000	3000
Grube Renate Ilse Bergbau A. G. Germany	1550 (110)	880 (470)	36 (2.5)	185 (13)	—	12,000	3000
Dow Chemical Co., Midland, Mich. U. S. A.	1400 (100)	700 (370)	360 (25.3)	360 (25.3)	—	3700	3600
Witkowitz, Czecho-Slovakia	1700 (120)	915 (490)	Cond.	200 (14)		18,000	3000
„	1690 (119)	930 (500)	Cond.	192 (13.5)	3	36,000	3000
English Electric, Rugby, England	(3200)/1400 [(225)/100]	842 (450)	200 (14)	—	—	360	25,000
Siemens-Schuckert, Berlin, Germany	(3200)/1700 [(225)/120]	760 (405)	185 (13)	—	—	1000	10,000
„	(3200)/2560 [(225)/180]	790 (420)	85 (6)	483 (34)	—	3200	6000
Langerbrugge, Belgium	(3200)/2800 [(225)/200]	842 (450)	710 (50)	—	—	4000	7500

in operation and on order

Number of Casings	Number of Sets	Operating since	Maker	Publications	Sectional Arrangement in Fig.
2	1	1930	*B. T. H.*	Power 69 (1929) pp. 909/10	—
2	1	1927	*Escher-Wyss*	*Escher-Wyss*-Mitt. 1928 pp. 121/28	—
2	1	1924	*B. B. C.*	Z. VdI. 70 (1926) pp. 711/15	—
1	1	1928	„	*B. B. C.*-Mitt. 15 (1928) pp. 321/25	—
3	1	1929	„		143
			de Laval		—
1		1927	*Westinghouse*		—
1	1	1929	*English Electric*	Engineering 74 (1927) pp. 610 et seq.	233
1	1	1925	*Amer. G. E. C.*	Power 68 (1928) pp. 713/18	—
1	1	1929	„		232
1	1	on Order	„		—
1	2		*Westinghouse*		—
1	1	1926	*Amer. G. E. C.*	Power 66 (1926) p. 822 Engineering 75 (1928) pp. 25/28 and 55/58	—
1	1	1929	„	NELA-Serial Report on Steam Turbines Aug. 1929 p. 74	—
1	1	1929	*Westinghouse*	Power 69 (1929) p. 326 NELA-Serial Report on Steam Turbines Aug. 1929 p. 73	235
1	1	on Order	*Amer. G. E. C.*	Electrical World 1929 p. 919	—
2	2	„	„		—
2	2	„	„	Power 69 (1929) p. 846	—
2	1	„	„	Power 69 (1929) pp. 994/97	238
2	1	„	„	NELA-Serial Report on Steam Turbines Aug. 1929 p. 70	—
2	1	„	„	Power 69 (1929) p. 863	—
			de Laval		—
1	1	1926	„	Engineering 74 (1927) pp. 164/65 Arch. Wärmewirtschaft 9 (1928) pp. 46/47	231
1	1	1929	*B. B. C.*	*B. B. C.*-Nachr. 17 (1930) p. 3 Z. VdI. 73 (1929) p. 913 and 993	—
1	1	1929	„		236
3	2	1930	*A. E. G.*	Proceedings of the 4th meeting of the Commission on H.P. plants of the "Vereinigg. d. E. W." (1929) p. 71	234
1	1	on Order	*B. B. C.*	*B. B. C.*-Mitt. 17 (1930) p. 93	—
4	1	1928	*Erste Brünner*	Z. VdI. 71 (1927) p. 447	—
3	1	on Order	*B. B. C.*		—
1	1	1924	*English Electric*	Engineering 74 (1927) pp. 610 et seq.	—
1	1	1926	*Escher Wyss*	Z. VdI. 71 (1927) pp. 446 and 595	—
2	1	1927	„		—
1	1	on Order	*B. B. C.*	*B. B. C.*-Mitt. 17 (1930) pp. 39/43	—

V. Condensers and their auxiliaries

During the last ten years engineers have been striving to improve steam turbines by all scientific means and have had the success we know; the problems connected with condensation, however, have only been confronted in the last few years and at the present time they are hotly disputed. For this reason, it is not yet possible to give any general rules on the best design of condensers and in the following pages only brief references will be made to the different and sometimes contradictory opinions and the reasons for upholding them. These differences will appear when a comparison is made between a few recent condensers of well-known makes; at the same time, however, certain tendencies may be noted pointing towards a gradual uniformity of design which has only been established for questions of secondary importance. At the present time no single design can be considered unquestionably superior to any other. As, furthermore, only few comprehensive results of reliable tests taken by experts are available and a perfect method of comparing condensers of different types has not yet been found, it is extremely difficult to compare various designs.

A good condenser should not only be capable of maintaining a high vacuum and producing a clean condensate free from gases and at as high a temperature as possible, it should also have a high specific capacity or, in other words, it should require but little space and power. A high vacuum increases the heat drop available in the turbine; a hot condensate reduces the heat of evaporation in the boilers.

The vacuum which is theoretically obtainable is determined by the conditions of the supply of cooling water. The temperature cannot be altered, at all events this is the case with fresh water out of lakes or rivers, for instance, and it is practically true when the water is re-cooled. The only quantity which may be chosen freely to begin with is the amount of cooling water, but even here the price of piping and pumps, for example, impose certain limits. For inland conditions in Europe the limit value of the water ratio is between 50 and 70 at the present time. The temperature and the chosen quantity of cooling water determine the vacuum which is theoretically obtainable when condensing a given quantity of steam. This is illustrated in Fig. 3.

The velocity of the cooling water is a quantity which depends directly on the amount of cooling water and its value has a great influence on the heat exchange. Thus, the average coefficient of the transmission of heat through tubes increases approximately in proportion to the square root of the water velocity. On the other hand, the resistance increases approximately as the square of the velocity and for this reason the velocity of the cooling water is seldom chosen essentially higher than 6.5 ft./sec. (2 m./sec.). However, high velocities are an advantage in a plant which is in continuous operation as it scours the tubes and prevents deposits forming so quickly when the water is dirty.

In the case of an ideal condenser, the chief superiority of which would be in the elimination of air, the vacuum obtained would be that corresponding to the leaving temperature of the circulating water. In practice it is, naturally, never possible to obtain quite such a low pressure, or high vacuum, for several reasons. Firstly, a certain temperature drop is necessary for every heat flow and one will be required, in particular, for the transmission of heat from steam

to circulating water; secondly, every condenser has air leakages; thirdly, flow losses will occur in the condenser although they may be small; and lastly, it is often necessary to depart from the best arrangement of the tubes for practical reasons.

Vapour and air are intimately mixed. The steam condenses, therefore, at the temperature corresponding to its partial pressure. As the mixture of vapour and air penetrates between the tubes, the vapour condenses and the volume of air increases. Thus, the saturation temperature decreases as the partial pressure of the vapour diminishes, and the cooling surface near the air suction will have to be at a very low temperature. This will have the undesirable effect of cooling the condensate as not only will the water condensed in this region be cold, but the hot condensate from the higher tubes will be cooled when falling. For this reason, it has become common practice in the design of modern condensers to imagine the entire surface divided into two regions, the region for condensing the steam and the region for cooling, concentrating, and drying the air. The arrangement of the two surfaces inside the shell and the ratio between them are questions still awaiting scientific and general answers. In the meantime, they are settled according to the experience and opinions of each maker and they are the reason for the largest divergence in designs.

The losses due to the flow of the steam in the condenser depend, naturally, on the arrangement and spacing of tubes. All modern condensers have wide entrance spaces between the nests of tubes to facilitate the flow, their chief object, however, is to ensure as even a division as possible of the steam over the entire surface. The spacing is often reduced towards the bottom of the condenser. This arrangement enables also the velocity of the steam to be kept as constant as possible through the condenser, a condition which is desired by many makers. Former designs frequently had closely pitched tubes filling the entire shell, present designs have varying spacing and the tubes are grouped in separate nests.

There is still another reason for separating the tubes in some designs; it is the so-called regenerative effect (68). The object is to bring the falling condensate as much as possible into contact with the warm steam so as to avoid any undercooling. If there are certain places where the condensate does get cooled, it is drained in such a way that it comes into contact with steam entering the condenser and will be regeneratively heated to the steam temperature. Naturally, a condenser of this type will have so many separate nests of tubes that the resistance to the flow will be quite negligible.

The specific duty of a condenser, which is the amount of steam to be condensed per unit area of cooling surface, influences also the heat transmission. Its chief importance is its effect on the size and price of the condenser. The duty has been continually increased during the last few years and at the present time it is between 7 and 10 lb./sq. ft. (35 to 50 kg./m.^{2}) for ordinary conditions but condensers have been made for considerably higher duties.

Finally, the method of arranging the flow of water and steam has an influence on the design. All modern condensers are designed for counter or cross-flow. There is no arrangement of the water flow which has been generally adopted; there are condensers for one or for several flows. The two-flow type is now the most common; the water is reversed once and passes through the tubes twice. The water is cold in the first flow and is employed in the air cooling zone, when it is warmer it goes through the tubes near the exhaust inlet. In America condensers of large capacities are almost exclusively built for one flow and designs having three or more flows are not often used.

---

(68) See *Th. Petty:* "Regenerative surface condensers", The Engineer 147 (1929) pp. 229, 259, 313, 346, 388.

Fig. 239 shows an *A.E.G.* two-flow condenser with a typical arrangement of the tubes. The coldest cooling water flows through the lower tubes. Wide passages penetrating far into the nests of tubes provide a large area for the steam flow and an even distribution over the whole length of the condenser. Baffles are placed below the top nests of tubes to catch the water and prevent it falling on the lower tubes. Several branches are provided on either side for extracting the air and the air cooling regions are covered by plates for catching the condensate. There is no attempt to obtain any regenerative effect as mentioned above.

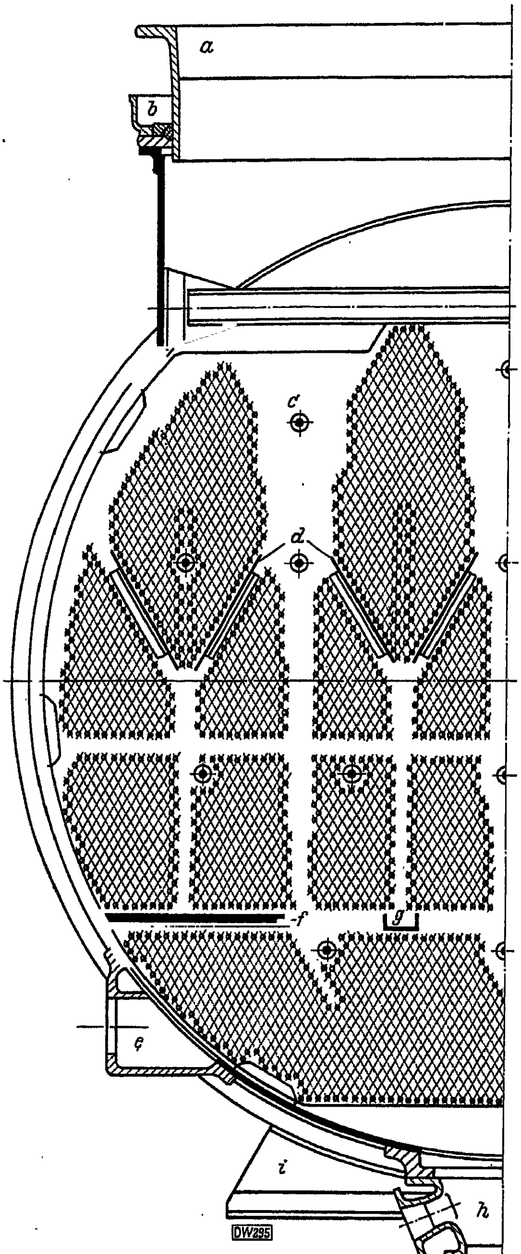


Fig. 239. *A.E.G.*, arrangement of tubes in a two-flow condenser

a = Expansion joint
b = Water seal
c = Stay bolt
d = Baffle plate
e = Air suction
f = Protecting plate
g = Collecting channel
h = Hot well
i = Condenser stool

A somewhat similar design is the *Delas-Ginabat* condenser in Fig. 240 as it does not provide for any regenerative effect either. It has also wide passages for admitting the steam between the nests of tubes. However, the advantages which are most frequently claimed for these condensers in the technical press are the following: firstly, the condensate draining off a tube in the centre of a nest falls tangentially on to the lower tube; secondly, in the direction of the steam flow the nests of tubes are very flat and most of the condensate can fall in passages between the nests of tubes. Thus, the condensate would be only slightly undercooled. In this example also, the air cooler is screened, but it seems to form only a very small proportion of the total cooling surface. Up to the present the superiority of this condenser has not been proved by any sufficiently accurate comparative tests.

The OV-condenser of *B.B.C.* (Fig. 241) is provided with a wedge-shaped space down the centre. It points downwards and allows the steam entering the condenser to distribute itself evenly and penetrate between the tubes almost in a horizontal direction. The air and non-condensable gases are extracted at both sides. According to the makers the condenser would have the following qualities: on account of the large entrance area the entire surface is equally effective; as a

result of the special arrangement of the tubes the velocity of the steam is almost constant and stagnant regions will not occur; the circulating water when passing through the different flows remains almost at the same level and undercooling is greatly reduced because the condensate only comes into contact with tubes at approximately the same temperature.

The *Escher-Wyss* condenser is of similar design (Fig. 242). The tubes, however, are arranged in nests which are separated by inclined baffle plates. The air is extracted at both sides through a screen.

In spite of the fact that these last two examples are not designed for obtaining any regenerative effect, they have many points in common with *Weir* regenerative condensers (Fig. 243). For instance, the right and left outer nests of tubes are for the first flow of the cold circulating water, whilst the inner

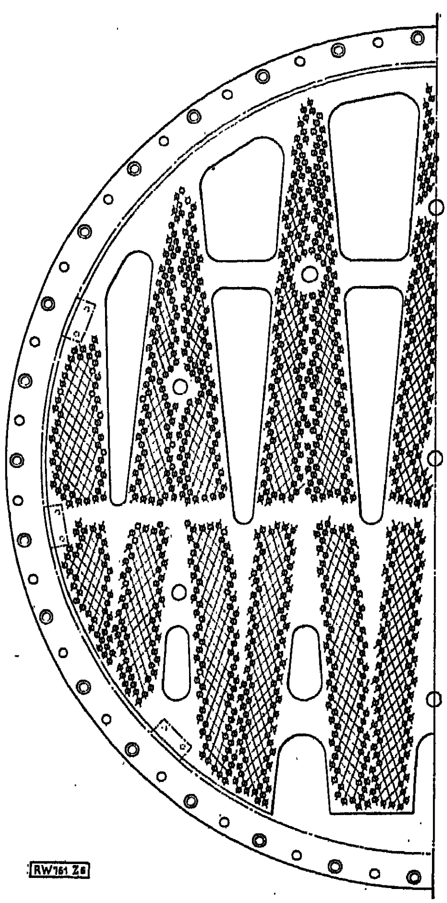


Fig. 240. *Delas-Ginabat*, arrangement of tubes in a two-flow condenser

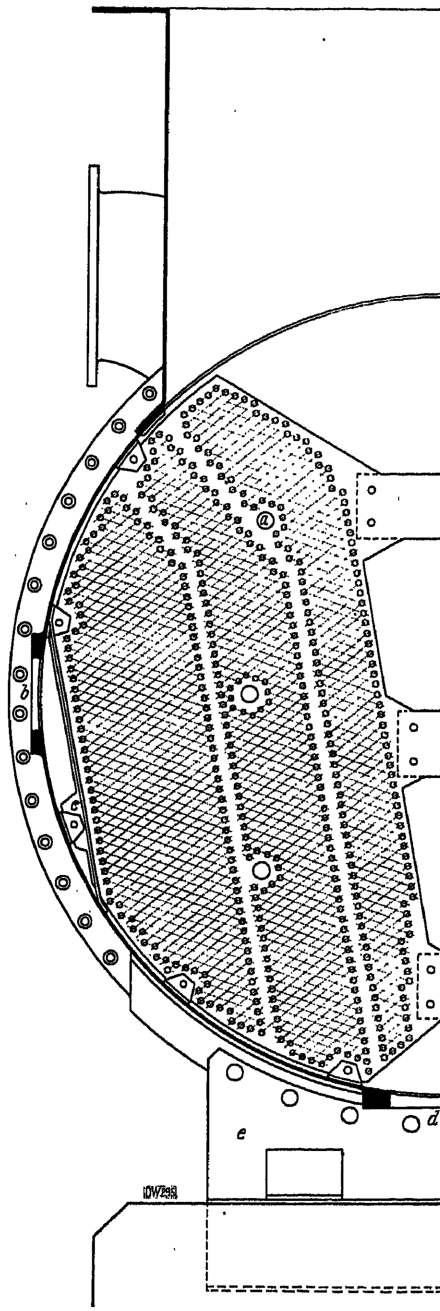


Fig. 241. *B.B.C.*, arrangement of tubes in a two-flow condenser

a = Stay bolt
b = Air suction
c = Baffle
d = Condensate suction flange
e = Condenser stool

nests of tubes are for the warmer water. A different reason is given, however, for the large central passage. The warm steam entering the condenser would come into close contact with the condensate falling from the nests of tubes and collecting at the bottom. An exchange of heat would take place and the condensate is always at approximately the same temperature as the warm steam. The very large air coolers on either side of the condenser should be noticed.

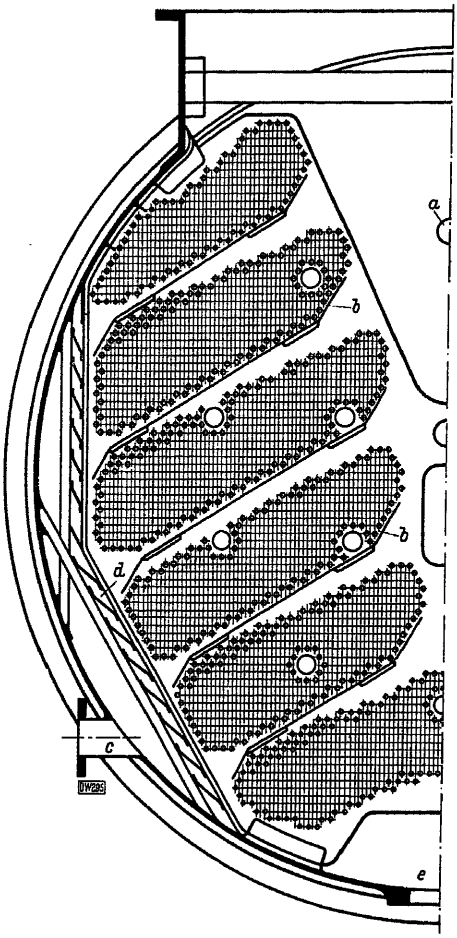


Fig. 242. *Escher Wyss*, arrangement of tubes in a two-flow condenser

a = Stay bolt
b = Baffle plate
c = Air suction
d = Baffle plates
e = Condensate suction flange

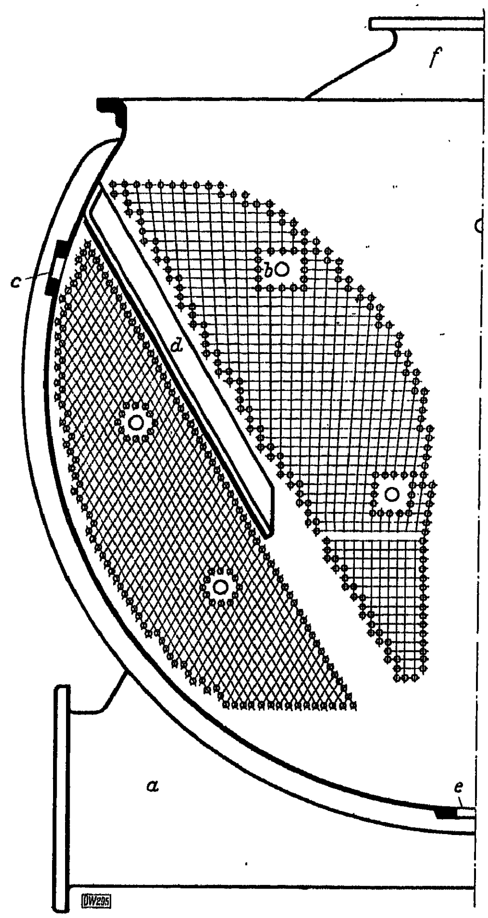


Fig. 243. *Weir*, arrangement of tubes in a two-flow condenser

a = Cooling water inlet
b = Stay bolt
c = Air suction
d = Baffle plate
e = Condensate suction flange
f = Cooling water outlet

They are well covered by baffle plates and even large leakages of air should have little effect on the vacuum. *Weir* also state that, contrary to what occurs in other designs, the difference in temperature between the steam entering the condenser and the condensate does not rise at partial loads and the temperature of the condensate will increase with the air leakages.

Whilst we are discussing condensers for which the advantage is claimed that condensate and exhaust steam are almost at the same temperature, a curious fact

may be mentioned. It was observed first on a condenser of this type but has since been noticed in a few other designs. It was found in certain cases that the temperature of the condensate was actually higher than that of the steam entering the condenser. It is true that in a number of instances errors in reading or faulty methods were discovered, but there remained some tests which appeared to be irrefutable. Attempts have been made to explain the phenomenon by means of the diffusor effect or by supposing that the steam is supersaturated on entering the condenser. If the second explanation is accepted, as is done by *Weir* especially, the occurrence would, naturally, be completely independent of the condenser design. It will be interesting to see the final explanation of a phenomenon which appears so contradictory (69).

Fig. 244 shows a *Weir* condenser for a marine turbine. It is designed according to the same principles as the stationary condenser which has just been mentioned and the incoming steam has direct access to the condensate collecting area. The condenser has two flows; the cooler water is used in the air cooler and the lower part of the condenser whilst the warmer water flows through the upper tubes. The steam which has not been condensed by the tubes of the first flow is carried along with the air and non-condensable gases and rises towards the air outlet. The condensate which is formed falls into a region at a higher temperature and any tubes with which it may come into contact will be warmer. It appears that with this condenser, also, the condensate was 1 to 2° F. (½ to 1° C.) warmer than the exhaust steam at the acceptance tests (70).

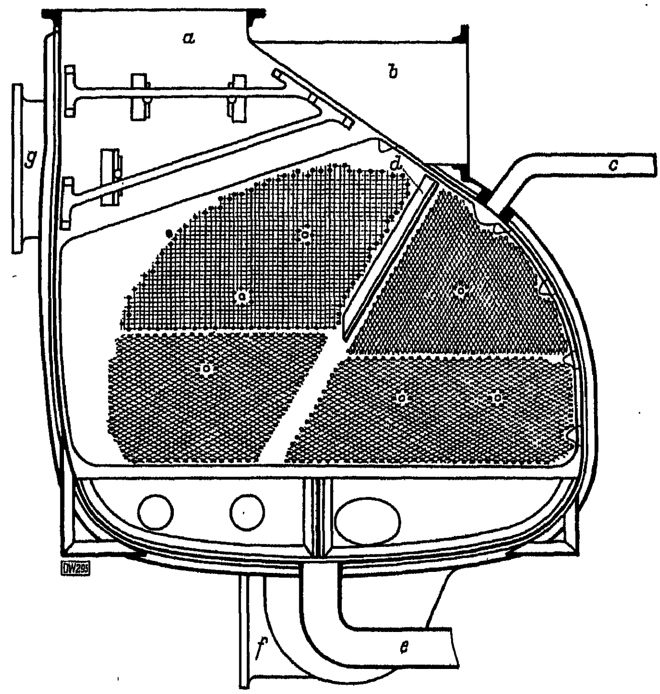


Fig. 244. *Weir*, arrangement of tubes in a two-flow marine condenser

a = Steam inlet from main turbine
b = Steam inlet from auxiliary machines
c = Air suction
d = Baffle plate
e = Condensate suction
f = Circulating water inlet
g = Circulating water outlet

The same principles as *Weir*'s have been adopted in America by the *Worthington Pump and Machinery Corporation* (Fig. 245). It may be noticed that the condensate flows through pipes across the entrance to the air cooling zone. The object is to reduce the surface of the condensate which is exposed to the comparatively cold draught of air and remaining vapour and to prevent it being carried along with the stream.

The tube bank in a *Westinghouse* condenser forms a very indented pattern (Fig. 246). The arrangement of a single-flow horizontal condenser is shown.

---

(69) See The Engineer 146 (1928) p. 489.
(70) See The Engineer 147 (1929) p. 273.

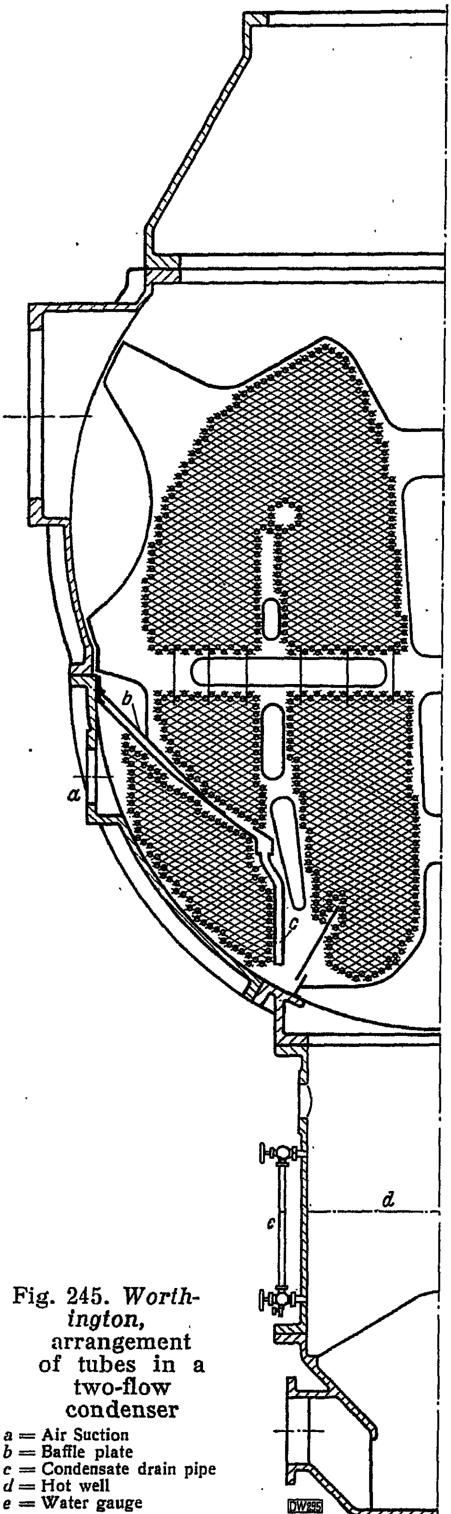


Fig. 245. *Worthington*, arrangement of tubes in a two-flow condenser

a = Air Suction
b = Baffle plate
c = Condensate drain pipe
d = Hot well
e = Water gauge

The condensate from each nest of tubes is caught by baffles, it falls in open spaces between the tubes and comes into contact with warm vapour. By missing the lower tubes it will not reduce their efficiency or become further cooled. The wide belt around the sides allows the steam direct access to the condensate collecting area. It may be noticed that a pipe along the axis of the condenser is used for the air suction. The lower section of tubes forms the air cooler; it is protected by baffle plates from the rain of condensate and, in this case also, the remaining mixture of air and vapour flows upwards.

Fig. 247 is a diagrammatic sketch of a vertical condenser of the same firm. It has a single flow and an original arrangement for the circulation of the water. The cooling water enters underneath, flows upwards through a large pipe in the centre of the condenser and passes down through the tubes. The air suction pipes are taken upwards through the central circulating water pipe, they are surrounded, therefore, by the coldest water and the cooling of the air will be very effective. Other details in designs of vertical condensers have already been mentioned (refer to p. 130).

The *Foster Wheeler Corporation* arrange their tubes along lines radiating from the bottom of the condenser (Fig. 248), thus obtaining a decrease in the pitch of the tubes and steam passages, which results in a constant velocity of vapour through the whole tube bank. In this example again, the condensate collecting area is in direct communication with the exhaust opening. The air coolers may be external, as in this case, or integral with the main condenser. The design shown has two flows but when the cooling water conditions make it advisable to have only a single flow, the same type can be built without any reversal of the circulating water.

Concerning details of design which are similar or the same in all types of condensers, reference should be made to text-books (71).

In order to obtain an even distribution of the steam over the whole condenser the

(71) Refer to *K. Höfer:* "Die Kondensation bei Dampfkraftmaschinen" (Berlin: J. Springer 1925) and *H. Balcke:* "Die Kondensatwirtschaft" (Munich: R. Oldenbourg 1927).

length of the tubes should not be chosen too great. The value will depend, naturally, on the shape of the exhaust connection. If there is only one circular exhaust opening in the middle of a long condenser, the tube surface at the top and at the ends will be of little use. This will not occur when the exhaust opening is very long, the limit being when the opening stretches over the entire tube length.

A satisfactory arrangement is obtained when the turbine has two exhaust branches (Fig. 249) as there will be a greater probability that the entire length of tube will be equally effective. The tendency in recent years has been to consider that a good condenser should be short and deep. Recently, however, some firms seem to have been favouring longer condensers again, the tube bank being, naturally, more shallow.

When the tubes are long—a large single-flow condenser in America with a cooling surface of 86,000 sq. ft. (8000 m.²)

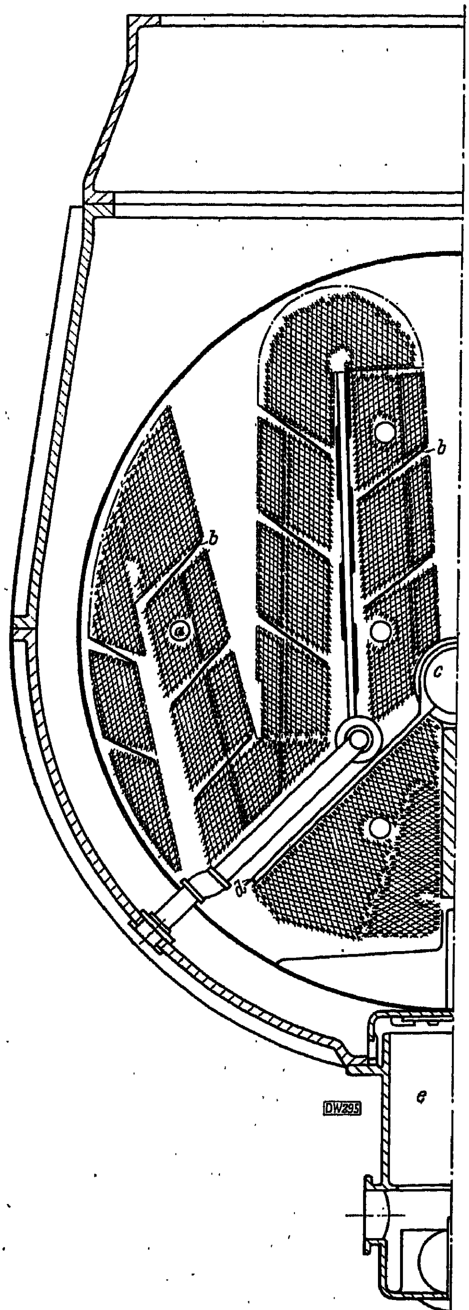


Fig. 246. *Westinghouse,* arrangement of tubes in a two-flow condenser

a = Stay bolt
b = Baffle plate
c = Air suction
d = Baffle plate
e = Hot well

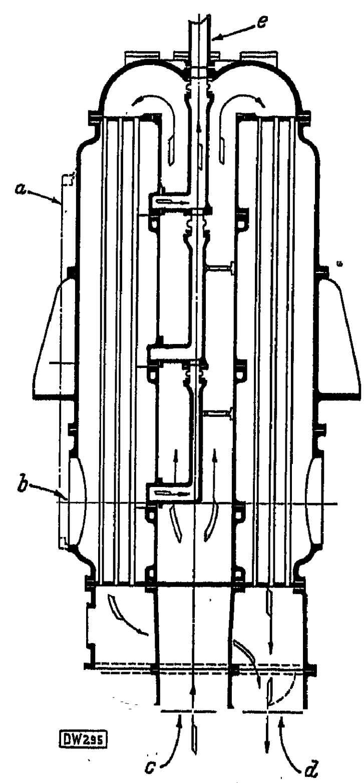


Fig. 247. *Westinghouse,* section through a single-flow vertical surface condenser

a = Steam inlet
b = Equalizing connection
c = Cooling water inlet
d = Cooling water outlet
e = Air Suction

has tubes nearly 26 ft. (8 m.) long—it is necessary to provide supporting plates to reduce the deflection and diminish the tendency to vibrations. Their arrangement should be carefully considered as they may prevent an even distribution of the steam over the entire length of the tubes. The *Ingersoll Rand Co.* have made a virtue of necessity; they design their plates as continuous partitions and each compartment has its own air and condensate outlet. This method will undoubtedly avoid all air pockets and ineffective portions of cooling surface.

It is highly important that the joint between tubes and tube plates should be perfectly tight. The best method of obtaining this result is still a matter of opinion. Some firms always provide stuffing boxes at both ends and use fixed tube plates. Others expand the tube at one end and have stuffing boxes at the other. In this case also, both tube plates may be fixed. Recently, however, the practice of expanding the tubes at both ends has been extending. In order to provide for the heat expansion of the tubes it will be necessary either to provide a stuffing box for one of the tube plates, which will then be free to slide inside the condenser shell, or the tubes must be capable of bending. Up to the present tubes expanded at both ends have proved most satisfactory, provided the expanded portions have been sufficiently annealed; these should not, however, be too soft as they will work loose. They will then have to be expanded afresh until they acquire sufficient hardness through the operation.

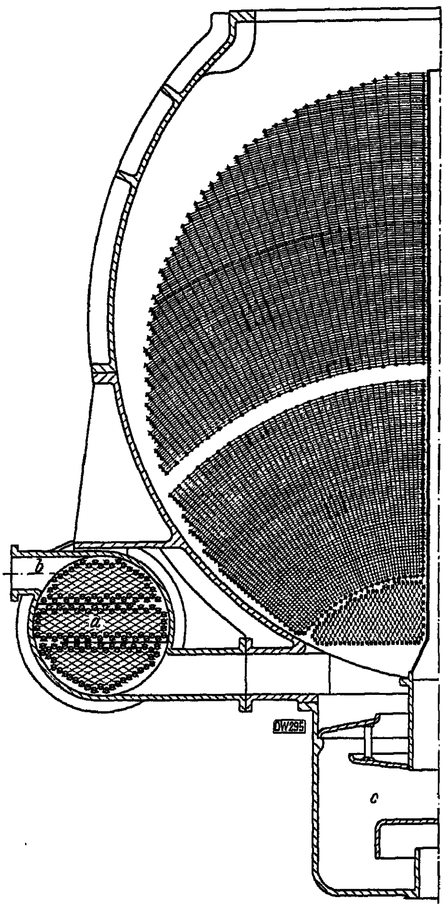


Fig. 248. *Foster Wheeler,* arrangement of tubes in a two-flow condenser

a = Air cooler b = Air suction flange c = Hot well

The packing should be renewed from time to time, especially if the condenser is sometimes overheated. The type of stuffing box which has proved most successful is the one with cotton cord packing as is generally used in marine condensers. Caulked metallic packings have been used also for some time. In order to obtain a perfectly tight joint with a metallic packing, however, the tubes must be held in the plates as firmly as if they were expanded, and they will have to bend in order to take up the heat expansion. Thus, there will be no advantage in using stuffing boxes.

Condensers for high specific duties are satisfactory only if they produce a high vacuum when in continuous operation; they should deliver condensate which is not contaminated by the circulating water and is sufficiently clean to be used as feed-water. If the guaranteed vacuum is to be maintained, the cooling water should be clean and not liable to produce scale. When it is not very clean, deposits of mud or scale may be diminished by increasing the tube velocity, this, however, will raise the power required for the circulating pump. Another method is a chemical treatment of the cooling water. There is very little advant-

age in dividing the water box and its cover in two parts so that each side may be cleaned separately; it is much better to do the operation as quickly as possible by means of gangs of cleaners working on the whole condenser at once.

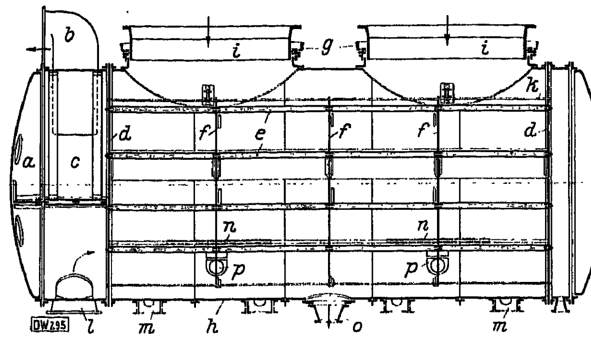


Fig. 249. *A.E.G.*, section through a two-flow condenser

a = Cover
b = Cooling water outlet
c = Water box
d = Tube plate
e = Stay bolt
f = Supporting diaphragm
g = Water seal
h = Condenser shell
i = Expansion piece
k = Tube
l = Cooling water inlet
m = Condenser stool
n = Baffle plate
o = Hot well
p = Air suction

A really effective method of protection against corrosion has not yet been found. None of the methods, such as the *Cumberland* protective system, the use of zinc plates or the tinning of tubes, has yet proved satisfactory. Erosion also occurs frequently and as it spreads very rapidly once it has started, it may have serious consequences. Reference should be made to text-books for theories on the cause of tube failures (72).

Condenser shells are made of plating or cast iron. The first named method is the cheaper and lighter. On the other hand, cast iron condensers may easily be made in irregular shapes, also their dimensions may be chosen more freely as stiffening ribs may be provided. Cast iron shells are very frequently used in America for large units, but most condenser makers in Europe favour plate shells.

For joining the condenser to the exhaust flange of the turbine either one of the numerous types of stuffing boxes is used (e. g. the water-sealed gland of the *A.E.G.* or the rubber expansion piece of *Westinghouse*) or the two flanges are rigidly bolted together. With this last arrangement, of course, the whole condenser must rest on springs.

Besides the condenser, a condensing plant will include also the auxiliary machinery for extracting the air and condensate, and the circulating pumps. Ejectors are generally used now for extracting the air and they may be of the steam or water jet types. Modern extraction pumps and circulating pumps are usually of the centrifugal type (73).

It would be possible to compare ejectors of the steam and water jet types by finding the energy required in each case for the same conditions. It is only on rare occasions, however, that this method is adopted. The results would never correspond to the actual service conditions. Not only is it impossible to arrive at conclusions by such methods, but, if technical publications are consulted, it will be found how uncertain and contradictory are the opinions on the subject. Whilst some declare that the superiority of steam operated ejectors has been established, others produce test results which are favourable to the water jet type. As has already been said, the operation of ejectors is still very little known and every design is based on personal opinions; thus, it often happens that tests with the same type of ejector give very different results. Very small variations in design may greatly alter the performance. It is often stated that water operated ejectors do not require the wet air to be cooled as much as steam

(72) See *O. Lasche-W. Kieser:* "Materials and design in turbo-generator plant" (London: Oliver and Boyd 1927).

(73) For examples of all kinds of condenser auxiliary machinery refer to *K. Höfer:* "Die Kondensation bei Dampfkraftmaschinen" (Berlin: J. Springer 1925); refer to the *A.E.G.-Mitteilungen* 1928 pp. 569 and 621 and 1929 pp. 13 and 55.

operated ejectors, and they are able to deal with a larger quantity of air when supplied with the same amount of power. This opinion is based on the assumption that the vapour which is always entrained with the air will be immediately condensed by the water jet and water operated ejectors will be less affected by moisture in the air. If the reduced power is the result of the diminution of volume the question arises: how much of the vapour is condensed during the very short time the jet takes to cross the suction chamber. The vapour might only be condensed in the diffuser, in which case the condensation of the moisture contained in the air would have no appreciable influence on the capacity of the ejector.

Other researches have shown that the steam jet type is just as good as the water jet type for extracting the air. From the point of view of power consumed, steam ejectors of the inter-condenser type may be considered most economical. Apart from the trifling amount of power required for the compression, the radiation losses and the small quantity of heat discharged with the gases, all the energy in the steam will remain in the cycle. Naturally, the utilization of other sources of power should not be impeded. It must be admitted that an ejector of the water jet type is much easier to construct than one of the steam jet type, but it cannot be said that the more complicated steam operated ejector has proved to be more difficult to operate. If the qualities of the two types of ejectors are compared, special attention being given to the operation, it will not be possible to say without reservation which type is better. It is not the saving of a little heat or a few kilowatt-hours which will determine the choice, but it will be solely the conditions of operation of the condenser or, speaking more generally, of the entire plant.

The auxiliary pumps consist of a circulating, an extraction pump and, if the ejector is water operated, another pump will be required. Whatever the method of drive, it is usually common to all auxiliary pumps in modern condensing plants unless separate drives are provided for special reasons, such as to economize space on ships. Apart from the fact that several small prime movers will be less efficient than a single large one, a condensing plant with a common drive will be the easiest to operate and the most reliable. The only governing which the pumps require is for their speed, there will be no difficulty, therefore, in satisfying all the working conditions together. The method of drive to be adopted—electric motor or steam turbine—will depend on the local conditions. An electric drive will have the lowest first cost and will usually be adopted if another source of power is available in the event of a stoppage of the main supply. When the ordinary power supply fails the motors will be automatically switched on the emergency supply. If there is no second source of current upon which it is possible to rely, an auxiliary steam turbine will be provided in case of an emergency. This is done in most stations for the supply of power to the public, and it enables the simpler and cheaper electric drive to be combined with the very reliable steam turbine drive. In normal operation the motor supplies the power but if it fails or the current is cut off, the steam turbine will automatically take the load and will maintain the condenser in operation. A modern set of auxiliary pumps of this type is shown in Fig. 250. The electric motor is directly coupled to the circulating pump, and drives the pump for the water operated ejector and the extraction pump through gears. These last two pumps are coupled to the same pinion; a steam turbine is coupled to another pinion which will drive the set when starting up or when the current fails. Another example of an auxiliary pump set is given in Fig. 213.

As auxiliary turbines are required only as stand-by they should be chosen for their low first cost rather than for their efficiency. For this reason they usually have a two or a three-row velocity wheel. When the motors are taking the load the wheels of the turbine rotate in a vacuum, they will be cooled by a

small quantity of steam and the turbine will require only very little power when running idle.

Condensing plants are often in duplicate in large turbo-units and two sets of pumps supply separate condensers. It will be possible to connect one set of pumps to the two condensers without interrupting the operation of the main set. Thus, one pump may be closed down at any time for overhaul or for other

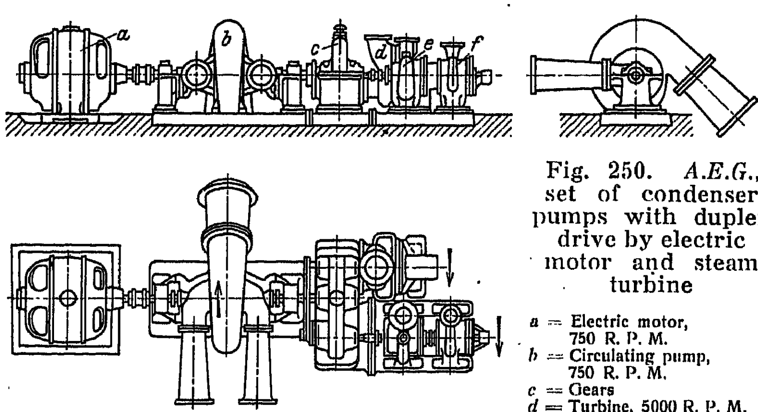


Fig. 250. *A.E.G.*, set of condenser pumps with duplex drive by electric motor and steam turbine

a = Electric motor, 750 R. P. M.
b = Circulating pump, 750 R. P. M.
c = Gears
d = Turbine, 5000 R. P. M.
e = Pump for water jet air ejector, 1500 R. P. M.
f = Extraction Pump, 1500 R P. M.

reasons. It is obvious that the vacuum will fall when only *one* set is in operation and a smaller load will be obtained for the same steam consumption.

The lay-out of a condensing plant will always be determined by the operating conditions. Any individual part of a plant is a member of a complete organism and has a definite task to perform. Thus, every condensing plant is a component part of a turbo-unit and of a thermal power station and has to be adapted to conditions upon which the entire operation will depend. It is only by co-ordinating the design of the plant and by amply rating the auxiliaries that a reliable operation will be assured.

Conclusion

A summary of the latest tendencies in steam turbine construction has been given in the foregoing pages.

The adoption of higher pressures and temperatures, the increasing output and speed, the resulting problems of appropriate materials and safe stresses, the many complicated methods of governing mixed-pressure and extraction turbines and, lastly, the endeavour to obtain the highest efficiency with the minimum of material, are all questions involving many problems which require profound study. Usually, turbines for high efficiencies demand the most careful designing, the most accurate workmanship and the best quality materials, and errors in design or manufacture will be of considerable importance.

Thanks to the untiring efforts of engineers, an unprecedented development has been accomplished in steam turbine design during less than a generation. It has had an enormous influence on power production in the civilized world and the capacities of power stations have risen to immense proportions. In spite of the great strides during the last few years, however, the competitive struggle for heat efficiency is not yet over and developments proceed. However different were the methods employed and however numerous are the applications of modern steam turbines, all efforts are directed to the same end: *the most reliable and economical prime mover*. The same object, the same considerations, the same experiences are leading to standardized forms. Variations in design must inevitably diminish as the solution of a definitely specified problem is approached.

Index

Accumulator steam turbine 147, governing 68, valve chest 39
Adjusting gear of turbine rotor 55, 62
A.E.G. 13, 24, 36, 39, 40, 43, 48, 52, 54, 56, 60, 67, 68, 69, 73, 104, 106, 111, 124, 135, 139, 143, 156, 170, 188, 195
Allis-Chalmers 43, 54, 73, 121, 123, 137, 141
Auxiliary machinery for condenser plant 195

Back-pressure turbine 9, 19, 20, 26, 131, efficiency at various loads 133
Baer 134
Balance piston 55
Balcke 192
Bauer 166, 169
Baumann, wetness of steam 7, multiple exhaust 41, 107, 121
B.B.C. 24, 34, 36, 42, 43, 44, 52, 55, 61, 65, 66, 67, 73, 104, 113, 114, 118, 121, 123, 125, 129, 135, 146, 160, 164, 177, 181, 182, 188
Bearing lining 87
Bearing pressure of gear teeth, permissible values 170
Bergmann 43, 52, 135
Blade fastening, dovetail 42, pine tree 43, serrated root 41, T-slot 42
Blowney & Warren 7, 14
Blue-brittleness 84
Boundary flow, effect on blade efficiency 26
Brass for condenser tubes 87
Bridging-over of the clearance *(B.B.C.)* 44
B.T.H. 47, 113, 120, 158, 180
By-product steam, by-product energy 19

Campbell 96, 118
Carbon-type packing gland 57
Carnot cycle 13, 14
Casing 46, exhaust 47
Cast-in plate blades 43
Cast iron 77, properties and composition 78
Cast steel 79
Chittenden & Law 180
Christlein 23, 24
Clausius-Rankine cycle 13, 14
Closing piece of a row of moving blades 43
Collingham 14
Combined governing 67
Compression shock 24
Condenser, load 187, vertical 52, 130, 192, water ratio 186
Condenser tubes, material 87, joint with tube plate 194
Corrugated pipe-type coupling of *Metro-Vick* 67
Coupling 67
Coverband, permissible stresses 94
Creep limit of material 88
Cumberland protective system for condenser tubes 195

Delas-Ginabat 188
Denecke 18
Diaphragm, brazed 36, built-up 34, cast 34, material and stresses 95, packing gland 56
Diffusing guide in exhaust casing 47
Diphenyloxide, reheating by means of 16
Disc friction loss 28, 31
Disc wheel, methods of mounting *A.E.G.* 65, *Amer. G.E.C.* 66, *B.B.C.* 65, *de Laval* 65, stresses 94
Dismounting device for disc-type rotors 65
Displacement work of blades 37
Double-extraction turbine, regulating system of *A.E.G.* 70
Double rotation turbine 149
Double toothed coupling of *A.E.G.* 172
Dresden 102
Drum, stresses 94

Efficiency, thermal 3, 4, thermodynamic 5
Ejectors 195
Elektra turbine 149, 152, 155
Elliott 101
"End tightening" method of *Parsons* 43
Endurance strength 88
English Electric 37, 43, 56, 110, 139, 179, 180
Erosion of moving blades 8, 90, of condenser tubes 90, 195
Erste Brünner 25, 35, 37, 43, 56, 106, 111, 114, 116, 141, 159, 175, 178
Escher Wyss 46, 72, 102, 109, 175, 189
Exhaust casing 47
Exhaust pressure, vacuum 4, 6, 9
Extraction back-pressure turbine 19, 143
Extraction diagram 138
Extraction turbine 19, 138

Fatigue limit of material 90
Feed-water heating 16
Fixed blades 34
Flow losses, effect of blade length 25, of steam velocity 23
Flügel 23
Ford Motor Co. 52
Forner 37, 177
Foster Wheeler condenser 192
Föttinger 155
v. Freudenreich 8

G.E.C., Amer. 29, 38, 56, 57, 66, 101, 117, 122, 129, 139, 157, 176, 178, 182
G.E.C., Brit. 43, 46, 52, 57, 75, 107, 109, 119
Geared turbine 155
Gears 161, loss 155, dimensions 170
Gerbel-Reutlinger 21
Glands 54, carbon 57, labyrinth 54, triple-flow labyrinth 56, water-sealed 57, interconnection 58
Governing 67
Growing of cast iron 78
Guard for stages with partial admission 39
Gümbel 59
Guy 14

Heat content 3
Heat drop 3, division over the parts of the turbine 5, gain with back-pressure turbines 9, gain due to improved vacuum 9
High-pressure turbine 101, division of heat drop 5, exhaust pressure 4, steam conditions 3
Hodgkinson 37, 101
Höfer 192, 195
Hort 96
House turbine 18, 145
Humboldt 30
Hydraulic test 91

Impulse method 33
Industrial turbine 19, 131
Ingersoll Rand condenser 194
Initial steam pressure 3
Initial steam temperature 4
Interstage gland 56
Interstage reheating 13, by means of diphenyloxide 16

Josse 23, 24, 125
Journal bearing, *A.E.G.* method of boring 63
Junggren 101

Keenan & Warren 24
Kerr & Mellanby 7, 14, 89
Kieser-Lasche 90, 195
Kingsbury 59, 62, 128
Kolb 152
Kraft 18
Krüger 83
Krupp, Izett steel 80, V5M steel 84
Kühnle, Kopp & Kausch 152

Labyrinth-type packing gland 54
Lacing wire 46
Lasche-Kieser 90, 195
de Laval 43, 65, 67, 135, 158, 177
Law & Chittenden 180
Leakage loss 25
Leaving loss 10, permissible value 12
Length of blades, effect on stage efficiency 25, relation to speed 29
Limit blade 39
Limit capacity turbine 12, 40, actually built by *A.E.G.* 13
Ljungström 56, 101, 149, 150, 151, 152, 155
Löffler 14
Lösel 25, 35
Low-pressure turbine 145
Lubrication, gears 173, generally 64, thrust bearings 60

M.A.N. 114
Manufacture of an *A.E.G.* blade 40
Marguerre 16
Marine exhaust steam turbine 168
Marine geared turbine 166
Martin 32
Material for blades, wheels, gears 82
Material, testing of the 87
Melan 14
Melchior 89
Mellanby & Kerr 7, 14, 89
Metro-Vick 41, 48, 55, 67, 107, 121, 165
Michell 59, 60
Mixed-pressure turbine 146
Monel Metal 83, 85
Moultrop 183
Mounting discs, method of *A.E.G.* 65, *Amer. G.E.C.* 66, *B.B.C.* 65, *de Laval* 65
Moving blades 33, 39, drop-forged 41, manufacture of an *A.E.G.* blade 40, material and permissible stresses 93, tapered and twisted 12
Multiple-exhaust *(Baumann)* 41, 107, 121
Multi-stage turbine 26, 27, 28, 31
Münzinger 14

Nickel-brass 87, 93
Nickel-chromium steel see stainless steel
Nickel steel 82, 93, 3% Ni 82, 5% Ni 83
Noack 14
Nozzle 34, governing 68, machined all over 35, tests 23
Number of stages 6, 23

Oerlikon 113
Oil governing of *B.B.C.* 74, of *Westinghouse* 75
Orrok 183
Outer packing gland 54
Overload nozzles 37

Pad-type single-collar thrust bearing 59
Parsons 28, 33, 42, 43, 55, 62, 63, 66, 104, 106, 107, 110, 113, 121, 158, 167
Parsons coefficient 31
Petty 187
Phosphorus, content in cast iron 78, in cast steel 79
Plate blade 39
Pochobradsky 109
Primary turbine 135, 163, 177
Principles of design 98

Quality figure 31

Rankine cycle 13
Rateau, accumulator 147, combined governing 67
Reaction method 33
Reheat, regainable heat 28
Regenerative effect in surface condensers 187, 190
Regenerative feed-water heating 16